Nominated Lecture

SMALL GAS TURBINES

By J. H. Weaving, B.Sc. (Eng.), Ph.D., Wh.Sch. (*Member*)*

INTRODUCTION

TEN YEARS AGO it was clear that in the aircraft field the gas turbine had established itself for military purposes, and even on the commercial side the reciprocating engine was fast becoming obsolete. It was, therefore, natural to examine how far the turbine invasion could be expected to penetrate into the automobile and small industrial fields. The advantage of light weight, paramount in the air, though desirable is less important for land applications, and there were many unknown factors and unanswered questions. What would happen to component efficiencies, particularly of compressors, in small sizes? What operational advantages and disadvantages does the turbine possess? And perhaps even more important commercially, how would the turbine compare in price with reciprocating units?

In this lecture it is intended to define small gas turbines as those not exceeding 500 hp and, although this is an Automobile Division lecture, it is nevertheless proposed to consider industrial as well as automobile turbines so that the subject of the lecture may be adequately covered and also because the problems with all small turbines are very similar.

Many of the problems could be assessed theoretically and it was immediately clear that thermal efficiency would be the greatest difficulty. For automobile applications a considerable advantage over the piston engine lay in the realization that a two-shaft turbine could offer excellent torque characteristics going a long way towards providing an automatic transmission; this, coupled with freedom from heavy vibration and absence of rubbing parts, provided a strong incentive. Nevertheless, only a limited assessment was possible theoretically owing to the many unknown factors, particularly that of the effect of scale on compressor and turbine efficiencies. For this reason many firms in many parts of the world felt that the only way of arriving at a satisfactory solution would be to make small experimental turbines. This was the view in Great Britain of the Rover Company and the Austin Motor Company on the automobile side, and, in the small and medium industrial field, Ruston and Hornsby and W. H. Allen and Co. In the United States of America, Chrysler and Boeing were first in the automobile field, with AiResearch and Solar for aircraft auxiliary and industrial applications. General Motors and Ford (United States) entered the race somewhat later but with great vigour and resources.

The general outcome, however, of some 10 years' development is that the turbine has failed to make the general conquest on land that it has done in the air, but that it has fitted admirably into certain applications where its characteristics are especially appropriate.

It is proposed, therefore, in this lecture first to look at the nature of the problem, secondly to consider the various approaches that have been made by the firms that have entered this field, and finally to examine in greater detail some of the problems of the small gas turbine that have been tackled by the Austin Motor Company, particularly in the direction of component efficiencies. It is believed that the experience of the author's company is typical of that of other firms.

Initially, most firms quite naturally wish to utilize to the full the basic simplicity of the turbine by using a simple cycle with as high a temperature as circumstances, such as price and available materials, will allow. Fuel consumption in most of the early units was very high but experience was not wasted as they demonstrated the possibilities of the gas turbine, in addition there are many industrial applications where fuel consumption is immaterial, such as emergency electric generating sets, fire pumps, etc. To meet applications where fuel consumption is important many companies have developed heat exchangers.

In order to make an assessment of the various approaches, it is desirable to examine the theoretical background. A simple cycle with and without heat exchanger will serve as a basis.

Typical component efficiencies for small gas turbines have been assumed and are given in Appendix I.

The temperature of 800°C (1472°F) has been selected as typical for industrial usage, where a reasonable life is required with a large measure of full-load running. Figs 1 and 2 show the effect of pressure ratio on thermal efficiency and

The MS. of this lecture was received at the Institution on 7th December 1961.

* *Chief Gas Turbine Engineer, Austin Motor Co. Ltd, Longbridge, Birmingham.*

fuel consumption respectively for full-load conditions, without heat exchanger and with a heat exchanger of thermal ratio varying from 0·6 to 0·9. These are considered to be the limits of practical heat-exchanger designs. Below 0·6 they would be scarcely justified and above 0·9 they become too heavy and bulky. The immediate points that emerge are that a heat exchanger is necessary for high thermal efficiency and, secondly, with a heat exchanger high pressure ratios are not required. With a 0·9 thermal ratio heat exchanger a pressure ratio of 3·5 gives a full-load efficiency of the order of that obtainable with a diesel engine.

It will be further observed that even with a 12/1 pressure ratio, where a heat exchanger shows no advantage on full load, thermal efficiencies are still unacceptable for most purposes. For part-load conditions the heat exchanger, however, would still have value but could scarcely be justified on cost as a two-stage centrifugal or multi-stage axial compressor would be required to obtain this pressure ratio.

With the same assumptions, using a 0·8 thermal ratio heat exchanger, Figs 3 and 4 show the improvement that could be gained with higher temperatures. These clearly demonstrate the incentive to develop materials for a turbine to accept temperatures of 1000°C (1832°F).

If it is wished to dispense with the heat exchanger, and most people who have developed heat exchangers would be very glad to do so, the alternative is high pressure ratios and temperatures. Calculations, however, show that a pressure ratio of 14/1 and a temperature of 1160°C (2120°F) would be required to obtain a thermal efficiency of 28 per cent with present component efficiencies. In calculating this temperature the compressor efficiency has been dropped to 76 per cent as it is believed that it would be impossible to obtain 80 per cent with present technique at this high ratio with small air flows. It will be clear, therefore, that for high efficiency a heat exchanger is unavoidable. At first sight it might be considered that for part loads a heat exchanger would be essential even with this high temperature but Fig. 5 shows there is little to choose between the moderate-temperature heat-exchanger unit and the very high temperature no-heat exchanger unit. Nevertheless, a heat exchanger would show an additional advantage under low-load conditions, even in the high temperature case, because with the lower speeds for part load, compressor outlet pressures and temperatures are lower and, if the compressor characteristic would allow the maintenance of a fairly high turbine inlet temperature, use could be made of the exhaust to heat the air after compression, as a sufficient temperature difference would then exist.

It will be appreciated from the above discussion that the selection of one system as being preferable to all others is impossible. As in most problems of engineering, the best solution to a particular application has to be decided as a compromise between several factors, in this case efficiency, reliability and cost being the principal ones. It is doubtless for this reason that the approach of various firms has been different; in many cases this is desirable as it has promoted competition between the different lines of approach, no doubt to the benefit of the development.

APPROACHES BY VARIOUS FIRMS

Table 1 has been compiled showing, in general, the latest engines of most of the companies which have produced small gas turbines, either experimentally or as production units. The two-shaft turbines are listed first. Most of these are for automobile purposes, though a group of engines are primarily for aircraft but many of these have been adapted to suit vehicles. These are followed by the single-shaft industrial machines.

Compressor and turbine efficiencies are maximum values, which do not usually correspond with the maximum pressure ratio. Some reservations should be made with regard to the weights quoted. These include reduction gear and accessories but these can differ very much for the different applications. The figures should therefore be only used as a guide. In the descriptions following, the engines are not described in detail but an endeavour has been made to highlight the main aim of the design and the unique features incorporated. It is hoped that with these short descriptions, illustrations, and Table 1, an adequate assessment of each unit can be made. Greater detail may be found from the references.

Austin Motor Company

The Austin Motor Company first looked at the possibility of small gas turbines, principally for automobiles, in 1949, when a design for a 120 hp unit was commenced with the sole idea of finding its potentialities. As the space available for a heat exchanger was not large it was decided to use a design which would give a fairly high pressure ratio if required. The unit is described fully in an A.S.M.E. paper (1)* and is shown in Fig. 6.

It consisted of a two-stage centrifugal compressor driven by a three-stage axial turbine followed by a single-stage power turbine. A single combustion chamber was used and a small matrix type cross-flow heat exchanger was incorporated to improve fuel consumption. The unit was installed in a modified 'Sheerline' car, which was tested in 1954. This experiment gave experience of the turbine in a motor vehicle and highlighted the advantages and disadvantages which are now well known. The advantages are smooth torque necessitating only a simple automatic transmission, freedom from heavy vibration, high specific power, no rubbing and wearing parts, easy starting, and the ability to use any distillate fuel. The main difficulties were high fuel consumption and the comparatively large amount of time needed to accelerate the gas generator making snap pick-up difficult. The experience, however, gave confidence that the advantages of the turbine made its development imperative, but at the same time showed that there was a long way to go.

A policy was therefore decided upon that would advance the development of the turbine but at the same time give prospects of a financial return in the near future, namely to

* *A numerical list of references is given in Appendix II.*

Table 1. Automobile and small industrial gas turbine data

Engine	Use	Type	Bhp	Compressor			Compressor turbine			Power turbine			Max. turbine inlet temp, °C	Max. turbine inlet temp, °F	Fuel consumption, lb/bhp.h	Combustion chamber	Heat exchange		Wt., lb	Spec. wt., lb/hp	Size L×B×Ht, in.
				Stages	Pressure ratio	Efficiency, per cent	Stages	Speed, rev/min	Efficiency, per cent	Stages	Max. speed, rev/min	Efficiency, per cent					Type	Thermal ratio			
Allen 350 kW .	Ind.	1 shaft	520	1-C	2·66	—	2-A	1500	—	—	—	—	700	1292	1·3	1 Rev.	No		2800	5·4	—
Allen 125 kW .	Ind.	1 shaft	185	1-C	2·5	—	1-R	23 000	—	—	—	—	700	1292	1·5	1 Rev.	No		600	3·25	39×34×50
Ruston . .	Ind.	1 shaft	430	1-C	3·5	80	3-A	19 250	84·5	—	—	—	760	1472	1·12	1 Rev.	No		3696	8·6	—
Auto Diesel .	Air-bleed	1 shaft	220	1-C	3·0	75	1-R	24 000	80	—	—	—	850	1562	1·46	2 Elb.	No		700*	3·2	47×37×34
Solar 'Mars' (Perkins)	Ind./Air	1 shaft	50	1-C	2·5	75	1-R	40 000	78	—	—	—	650	1202	—	1 Elb.	No		98	1·96	24·5×17×22
AiResearch GTP.30-1	Ind./Air	1 shaft	30	1-C	—	—	1-R	52 800	—	—	—	—	795	1463	1·4	1 Rev.	No		40	1·33	16×16×16 (3)
AiResearch GTCP.85-91	Ind./Air, Air-bleed	1 shaft	200	2-C	—	—	1-R		—	—	—	—	—	—	1·4	1 Rev.	No		260	1·3	28×33×33
Rover 1S/60 .	Ind.	1 shaft	60	1-C	2·9	—	1-A	46 000	—	—	—	—	790	1454	1·45	1 Rev.	No		133	2·2	19×18×24
Rover 1S/90 .	Ind.	1 shaft	90	1-C	3·0	—	1-A	46 000	—	—	—	—	790	1454	1·38	1 Rev.	No		133	1·5	19×18×24
Budworth 'Brill'	Ind.	1 shaft	90	1-C	2·8	—	1-R	41 000	—	—	—	—	807	1485	1·30	1 Ann. Mk. 2	No		115	1·3	28×16×16
B.M.W. . .	Ind.	1 shaft	50	1-C	3·0	—	1-R	45 000	—	—	—	—	—	—	—	1 Ann.	No		—	—	—
Deutz . .	Ind.	1 shaft	80	1-C	2·7	—	1-R	50 000	—	—	—	—	780	1436	1·32		No		139	1·73	30×20×25

* No reduction gear.
(3) Approximate.
(4) Including transmission.

USES

Ind. Industrial.
Air Aircraft.
Auto. Automobile.

COMPRESSOR AND TURBINE

C Centrifugal.
A Axial.
R Inward flow radial.

COMBUSTION CHAMBERS

Ann. Annular.
Str. Straight through flow.
Rev. Reverse.
Elb. Elbow.
Ann. Annular.

HEAT EXCHANGERS

Recup. Recuperative.
Regen. Regenerative.
L. Length.
B. Breadth.
Ht. Height.

Table 1—continued

Engine	Use	Type	Bhp	Compressor			Compressor turbine			Power turbine			Max. turbine inlet temp, °C	Max. turbine inlet temp, °F	Fuel consumption, lb/bhp.h	Combustion chamber	Heat exchange		Wt., lb	Spec. wt., lb/hp	Size L×B×Ht, in.
				Stages	Pressure ratio	Efficiency, per cent	Stages	Speed, rev/min	Efficiency, per cent	Stages	Max. speed, rev/min	Efficiency, per cent					Type	Thermal ratio			
Austin 120	Auto.	2 shaft	120	2-C	4	76	3-A	23 000	83	1-A	22 000	85	800	1472	0·8	1 Elb.	2 Recup.	0·65			
Austin 30	Auto./Ind.	2 shaft	30	1-C	3	77	1-R	56 000		2-A	24 000		800	1472	0·79 (est)	1 Str.	1 Recup.	0·65			30×34·5×28·5
Rover 2S/140	Auto.	2 shaft	140	1-C	4·0	80	1-R	65 000		1-A	50 000		927	1700	0·59	1 Rev.	2 Recup.	0·78			58×29×26 (4)
Chrysler CR2A	Auto.	2 shaft	140	1-C	4	80	1-A	44 610	87	1-A	45 730	84	927	1700	0·51	1 Rev.	1 disc Regen.	0·9	450	3·2	36×35×27
Ford	Auto.	L.P.H.P. power, shaft	300	L.P. 1-C, H.P. 1-C	16	80 80	L.P. 2-A H.P. 1-R	46 000 91 000	85 83	1-A	37 500	85	930	1706		2 Str.	Recup.	0·75	650	2·17	38×29×28
G.M.	Auto.	2 shaft	225	1-C	3·3	78	1-A	33 000	—	1-A	24 000	—	900	1652	0·59	2	2 Ann. Regen.	0·86	590	2·7	37×31×24
Boeing 502-10MA	Auto./Air	2 shaft	300	1-C	4·5	76†	1-A	37 500	82†	1-A	29 400	—	900	1652	0·92	2 Str.	No	—	335	1·12	42×24×24
Boeing 520-3	Auto./Air Ind.	2 shaft	500	1-C	6·6	—	1-R	44 500	—	1-A	27 000	—	882	1620	0·65	2 Rev.	No	—	340	0·68	64×25×26
AiResearch 331	Auto./Air Ind.	2 shaft	300/400	2-C	—	—	2-A	—	—	1-A	—	—	—	—	0·46 (est.)	1 Rev.	2 Recup.	—	—		52×36×30
		1 shaft	400	2-C	6·1	—	2-A	—	—	—	—	—	955	1750	0·66	1 Rev.	No	—	200	0·5	36×18×18
Blackburn Turmo 603	Air/Ind.	2 shaft	425	1-C	4	79	1-A	34 000	84	1-A	35 000	78	876	1609	1·1	1 Ann.	No		313	0·74	48×27×20
Fiat 8001	Auto.	2 shaft	290	2-C	7		2-A	30 500		1-A	29 000		800	1472		2 Str.	No		600	2·1	
Daimler-Benz	Auto.	2 shaft	300	1-A, 1-C			2-A	28 000		1-A	21 000		800	1472		1	No				
Austin 250	Ind.	1 shaft	250	1-C	3·5	80	2-A	29 200					800	1472	1·15	1 Elb.	No		900	3·6	63×41×51

L.P. Low pressure. H.P. High pressure.

† Efficiency at maximum speed, not maximum efficiency.

(See also notes below first part of table on p. 5.)

fields of operation where the turbine had obvious advantages. Fuel consumption being the major difficulty, emergency power applications seemed to be a first objective, and a 250 bhp industrial gas turbine, described later, was designed for this purpose. The second stage in this development, now well advanced, will be the addition of a heat exchanger and finally it is proposed to modify the unit by adding a free power turbine for traction purposes, heavy commercial vehicles, railcars, and off-road vehicles being visualized.

The Company's next two-shaft turbine suitable for vehicle use was a most interesting one, developed under contract for the Directorate of Industrial Gas Turbines of the Ministry of Supply*.

The contract was awarded to encourage Industry to investigate the possibilities of very small gas turbine units and this is undoubtedly the smallest two-shaft complete unit suitable for vehicle and similar use. To obtain reasonable fuel consumption a heat exchanger was a necessity and the greatest unknown factor was that of component efficiencies of such small units. Estimates were made, perhaps somewhat optimistically, and the design target shown in Table 2 was evolved. The unit consists of a single-stage centrifugal compressor driven by an inward-flow radial turbine followed by a two-stage axial turbine (Figs 7 and 8). The idea of using two stages was to minimize the difficulty of diffusing and ducting from the comparatively small eye of the exducer of the radial turbine to the inlet annulus of an axial turbine. The outside diameter of each axial wheel is approximately 5 in. and that of the radial turbine 5·6 in. To have had a radial power turbine would have involved even more complicated ducting with additional losses and also the probable need of having the shafts of the turbines at right angles if a compact design were to be achieved.

The unit, after some development, realized its 30 hp at

Table 2. Austin 30 hp gas turbine, design target

Power output	30 hp
Gas inlet temperature	800°C (1472°F)
Mass flow	0·85 lb/sec
Fuel consumption	0·79 lb/bhp.h
Compressor (single-stage centrifugal)	
Pressure ratio	3·0
Adiabatic efficiency	74 per cent
Compressor turbine (single-stage inward flow radial)	
Efficiency	80 per cent
Pressure ratio	2·08
Power turbine (two-stage axial)	
Adiabatic efficiency	75 per cent
Pressure ratio	1·34
Heat exchanger (cross-flow recuperator)	
Thermal ratio	0·7
Pressure drop air- and gas-side	4·5 per cent
Combustion chamber	
Efficiency	98 per cent
Pressure drop	4 per cent
Reduction gear	
Ratio	6·95/1
Output speed	3600 rev/min
Efficiency	95 per cent

* *Now the Ministry of Aviation.*

the design speed of 56 000 rev/min, but the target for fuel consumption proved more difficult, largely due to leakage in the cross-flow heat exchanger and failure to obtain the design efficiency with the power turbine with a possible slight mismatch between the power turbine and the inward flow radial compressor turbine. Although the contract has now been completed, further development has continued and these difficulties have now been largely overcome, a fuel consumption of 0·85 lb/bhp.h being obtained. This is not, of course, as good as its reciprocating counterpart, but the heat exchanger is of only moderate size and developments to increase the inlet temperature could well be made now the design has proved to be mechanically sound. However, price determinations have been estimated, and even with large numbers, such as 1000 per week, it is found that the unit must be cheaper by at least 50 per cent to make it competitive with its piston counterpart. Clearly this can only be made possible with intensive development in the field of design, production and materials. Until this has been achieved, in the author's opinion, the turbine is not an economic proposition for the small automobile. Nevertheless, the contract has shown that such a unit technically is quite a practical proposition and the challenge to cheapen the design is still most interesting, though formidable.

Rover Company

The Rover Company made newspaper headlines on being the first to demonstrate a vehicle powered by a gas turbine in 1950. This company was chosen by the Ministry to build some of the first prototype Whittle jet engines and thus has had a long background of gas turbine experience. In October 1961 (2) they announced their latest passenger car saloon with the new 2S/140 turbine engine (Fig. 9).

This latest Rover unit differs from the early models in the adoption of an inward-flow radial turbine to drive the compressor. The turbine is a precision casting mounted back-to-back with the compressor. The gases are ducted from the eye of the radial turbine to a single-stage axial-power turbine, which drives the transmission through an epicyclic forward/reverse gearbox.

The two heat exchangers are recuperators of the primary surface type and give a thermal ratio of approximately 78 per cent at full load resulting in a very creditable fuel consumption under this condition of 0·59 lb/bhp.h.

The Rover Company has made great efforts to minimize the difficulty of the acceleration of the gas generator section of the turbine, and to do this they have arranged for a high idling speed and to reduce the fuel consumption under these conditions they have made provision to reduce the swallowing capacity of the turbine. This has been cleverly accomplished by pre-swirl vanes at the entrance of the compressor which are pivoted to give two positions, one for full-load running, and the other to throttle the compressor for idling. In like manner the compressor turbine is matched by two-position nozzles feeding the inward-flow radial turbine; the nozzles of the axial turbine are not movable. The idling speed is 40 000 rev/min and the speed has to be increased

to 65 000 rev/min for the full rated power. In order to accelerate from idling the turbine is opened up to its full area position to give maximum speed and torque.

Chrysler Corporation

Chrysler (3) was the first American company to produce an automobile gas turbine, starting serious investigation in 1947. They have always realized the necessity for a heat exchanger, and were the first to adopt the Ritz type of rotary regenerator with which they have made considerable progress in development. Their theme has been the simplest unit consistent with efficiency and thus they have adopted a layout of a single-stage centrifugal compressor, one axial compressor turbine rotor, one axial power turbine rotor, one combustion chamber, and one heat exchanger. Clearly, no further reductions are feasible without going to a single-shaft unit, which would need a complicated transmission to be suitable for vehicles.

A 120 hp engine of this basic design powered the first car across the American continent from the Atlantic to the Pacific in 1956.

Their latest engine, the CR2A (4), is the third design produced over a long period of extensive component development and it has been exhibited in a futuristic car, the 'Turboflite', in the 1961 Paris, London and Turin Motor Shows. This unit follows the basic design of its predecessors but increased efficiency has been achieved in all the major components and, in addition, it accepts a slightly higher gas temperature of 920°C (1688°F). It will be noted that the layout (Fig. 10) is a compact one with a minimum of ducting. A unique feature to increase the efficiency of the axial power turbine over its large operating range of speed is the incorporation of movable nozzle blades to feed this rotor. Turbine blade angles are correct only if the ratio of gas velocity and blade velocity remains constant. In automobile applications it is desirable to be able to run the power turbine from stall to its maximum speed so that the transmission system is simplified, the only requirement being a reduction gear and a forward/reverse gearbox. However, although turbine blades will accept a certain amount of negative and positive incidence with only small loss of efficiency, clearly at very low speeds there is a considerable divergence from the optimum and consequently heavy loss of efficiency. One method of mitigating this condition is to use a two-speed or multi-speed transmission thereby limiting the normal range of power turbine speeds, except for the starting condition. Chrysler has designed to obtain the optimum conditions without change-speed gears by pivoting the nozzles feeding the axial-power turbine. There are 23 blades in the variable pitch assembly and they are set automatically to meet the varying conditions; by turning the nozzle blades to an extreme position engine braking is also obtainable. The action of opening up the nozzles from a predetermined position will increase the area of flow and thus the pressure drop, and hence the work, in the compressor turbine; this will improve the acceleration of the gas generator and reduce the time for making sufficient power available to the power turbine for quick vehicle acceleration. Additionally, throttling the output reduces the swallowing capacity of the unit and helps to maintain higher gas temperature at lower loads and can thereby improve part-load efficiency. However, many of these improvements are mutually exclusive and though little detail has been published doubtless a compromise has had to be made to obtain the best overall improvement. For instance, pivoting the nozzle blades while improving the incidence of the rotating blades is liable to produce worse conditions of incidence on the leading edge of the nozzles if a large angular movement is required. Again, throttling the outlet of the gas generator is limited as it will ultimately cause the compressor to surge. The Chrysler Company is, nevertheless, to be complimented on taking a bold step.

On the heat exchanger side the development of a regenerator of 90 per cent thermal ratio with a 4 per cent leakage is a good achievement if it can be sustained for a satisfactory life.

Ford Motor Company (United States of America)

The National Gas Turbine Establishment (5) performed a series of performance calculations considering the various ways in which the components of multi-stage gas turbines could be arranged. From these calculations it can be seen that a three-shaft engine with separate low and high pressure sections, heat exchanger, reheat, and intercooling, would be suitable for vehicle operation. This, however, seemed to lose the major advantage of the gas turbine, namely simplicity. Subsequently, however, the Ford Motor Company after making their own calculations (6) decided that the probable gains in fuel consumption would outweigh the complications and they are to be complimented on producing prototypes of what is perhaps the most interesting small turbine yet produced (7). Calculations showed that a considerable advantage in part-load economy could be achieved by having the power turbine as an intermediate stage between high-pressure and low-pressure wheels. Calculated fuel consumption curves are shown in Fig. 11 for two positions of power turbine together with a conventional heat-exchanger layout. The unit consists of two separate spools and may be considered as a turbocharged gas turbine. The low-pressure spool consists of a single-stage centrifugal compressor driven by a two-stage axial turbine, 'supercharging' a small single-stage centrifugal compressor cast integrally with an inward-flow radial turbine and run at 91 800 rev/min. Intercooling is provided between the compressor stages to reduce the power requirements. After compression the air passes through the contra-flow heat exchanger of the recuperative type and then to the first combustion chamber. After passing through the inward-flow radial turbine the gases enter a second combustion chamber where they are reheated to the same inlet temperature as the high-pressure turbine before entering the power turbine. The power turbine in turn discharges to the low-pressure turbine driving the first-stage centrifugal compressor.

There are no nozzles between these two turbines, which rotate in opposite directions. The intercooler is cooled by

atmospheric air which is drawn through the matrix by an engine-driven axial fan.

The performance so far published states that 247 hp has been attained at a turbine inlet temperature of 930°C (1706°F).

The turbine unit is a masterpiece of compact economical design as will be seen from Fig. 12 and the dimensions and weight in Table 1.

General Motors Corporation

General Motors joined the race in 1953 with a simple 300 hp vehicle unit without heat exchanger, from which they have developed, through one or two 'marks', their present GMT/305 turbine with rotary regenerator. A few of these units have been produced by the Allison Division as prototypes for various uses in a variety of different fields, from motor boats to off-road trucks (8).

The unit (9) consists of a single-stage centrifugal compressor with channel-type diffusers, driven by a single-stage axial turbine followed by another single-stage power turbine on a separate shaft which drives the transmission through the reduction gear (Fig. 13).

The outstanding features of the General Motors turbine are the high inlet temperature and regenerative heat exchanger. The high temperature is made possible by having extended roots and fir-tree fixings, as in aircraft practice, with disc cooling. In spite of this effect some blade fatigue failures were experienced in prototype testing (8).

The mechanical construction is outstanding in the almost complete elimination of ducting achieved by allowing the compressor to discharge into a plenum chamber which houses the two rotary regenerator drums. These are built up of layers of porous material alternating with thin metal plates, the whole being contained between two solid rims in the form of two annuli. Two 'window-frame'-type seals divide the drums into arcs of $\frac{1}{3}$ and $\frac{2}{3}$ of the circumference for compressor inlet-air and exhaust-gas discharge respectively. The outlet air after passing through to the inside of the drum flows directly into the combustion chamber without ducting and thence to the compressor turbine. The turbine exhaust flows into the low-pressure side of the heat exchanger and radially outward through the matrix to the exhaust. The drums are driven at approximately 1/1000 of the turbine speed. The seals are movably mounted and positioned by two rollers on each regenerator rim, thereby allowing drum expansion while maintaining a constant clearance. General Motors claim a clearance leakage of less than 5 per cent. The present thermal ratio is 0·86 but they have units of 0·9 ratio on test. Fuel consumption is shown in Fig. 14.

This compact design though commendable in many respects does make diffusion after the power turbine virtually impossible and thus there is a loss of kinetic head. This is to some extent compensated by using a low axial velocity in the design.

Boeing Airplane Company

The Boeing Airplane Company was early in the field with a two-shaft automobile turbine which they installed in a commercial vehicle with a conspicuous measure of success and, though unable to equal diesel engine fuel consumptions, they do claim that overall running costs need be no greater (10). The latest vehicle application has been to propel and supply power for pumping units for two fire-fighting vehicles, one for Seattle and one for San Francisco. Their philosophy of simplicity and light weight has been adopted, no doubt, because they have been concerned mainly with aircraft applications. In this connection Boeing units have powered a large number of aircraft and helicopters with conspicuous success, including the setting up of an altitude record for this class of turbine of 37 063 ft in 1953. The aircraft applications, with the philosophy of light weight, have prohibited the use of a heat exchanger and, to obtain an acceptable fuel consumption, Boeing have done considerable research into increasing the pressure ratio obtainable from the single-stage centrifugal compressor. The 502 series has a pressure ratio of 4·5, the compressor being driven by a single-stage axial turbine and feeding a second free single-stage axial turbine through two combustion chambers.

Very considerable development, encouraged by the United States Government, is to be seen in the latest 520 series (11), where an even more remarkable achievement in pressure ratio for a single-stage centrifugal compressor has been realized with a ratio of 6·5. Stresses at this peripheral speed are very high and to obtain a symmetrical stress distribution a double-sided impeller is employed, similar to a Whittle compressor.

An inward-flow radial turbine drives the compressor (Fig. 15), and this is one of the largest turbines of this type in production. The free power turbine is axial as in the 502 series.

The rotor assembly is stiff and runs below its first critical speed; it is supported by three slipper-type journal bearings and a slipper thrust bearing. These bearings, which build up a hydrodynamic oil film as in a Michell thrust bearing, are a special feature developed by Boeing and preferred to the more usual ball-and-roller types. In this design the combustion chambers have been brought closer together, allowing a short cross-fire tube. The whole unit is very compact and has a small frontal area, making it very suitable for aircraft. Industrial and vehicle applications, however, are also contemplated.

AiResearch

The model 331 gas turbine (Fig. 16) is the first vehicle turbine to issue from this company, which has had vast experience in light-weight auxiliary units (see below). It is also the latest automobile unit to be announced from the United States of America. Although full details have not been published it appears to be a good example of the compromise between a high-pressure-ratio no-heat-exchanger unit and the low-pressure-ratio heat-exchanger unit. It uses a moderate ratio of perhaps 8/1 obtainable with a two-stage centrifugal compressor driven by a two-stage axial compressor turbine followed by a single-stage axial-power

turbine. The air after leaving the compressor makes a double pass through a cross-flow matrix-type recuperative heat exchanger, the exhaust side of which is single pass. The estimated fuel consumption of 0·5 lb/bhp.h at full load is quite good. The unit is also to be made available without a heat exchanger. In this form the fuel consumption is 0·6 lb/bhp.h at full load, which indicates high turbine inlet temperature and component efficiencies. Part-load performance will, however, fall off without the heat exchanger. In the model 431, also recently announced, the two-stage compressor is driven by an inward-flow radial turbine with a second (free) radial turbine for output power. This most interesting unit has variable-area power-turbine nozzles giving improved part-load operation and a reverse without gears.

Blackburn Engines Ltd

Blackburn Engines Ltd manufacture a series of engines under licence from Turbomeca, France (12). These are largely used for light aircraft and helicopters and as air-bleed turbines for starting large jet engines. The 'Turmo 603' is the most applicable engine for consideration in this lecture as it is a two-shaft machine and is available for industrial purposes. However, all the units follow the same basic design and are of light compact construction (Fig. 17).

The stiff rotor assembly is worthy of note, as is also the centrifugal compressor with its long inlet guide vanes, which has been the subject of considerable development. One of its unique features is the annular combustion chamber, which is both short and contained within the contour of the centrifugal compressor, and this maintains the frontal area at a minimum, an important factor for aircraft usage. Also unique is the fuel injection system. Fuel, fed from the front of the engine through the auxiliary gearbox, passes along the axis of the compressor rotor shaft until it reaches the plane of the annular combustion chamber where it enters an assembly inside the large-diameter rotor shaft and is finally centrifuged out through fine holes drilled in this component. It will be appreciated that combustion differs from that in any other type of chamber in that the flame is truly annular and spreads very little in the axial direction.

Fiat Motor Company

Fiat Motor Company (13) have made an experimental automobile turbine of 290 hp and have adopted a policy, similar to that of Boeing, of endeavouring to obtain reasonable fuel consumption without heat exchanger. However, the pressure ratio of 7/1 has been obtained by a more conventional two-stage centrifugal compressor driven by a two-stage axial turbine. This has a weight penalty but, as it was not designed for air use, its weight is, in fact, by land standards very good. Characteristic features are the two parallel combustion chambers on top of the engine and the compact method of combining the transmission, including the differential axle drive, in the turbine casings.

Daimler-Benz

The little that has been published on the Daimler-Benz unit is shown in Table 1. It will be noticed that an axial stage of compression precedes the centrifugal compressor, no doubt to increase the pressure ratio a little for a no-heat exchanger unit. The fuel consumption quoted is surprisingly good for a turbine inlet temperature of 800°C (1472°F). There is also a heat exchanger version of this engine under development, of the rotary regenerative type. No figures, however, are published.

INDUSTRIAL GAS TURBINES

Examination is made the more interesting, albeit more complex, by the fact that in the range of 250 bhp and above no two designs are identical, even in the selection and arrangement of the major components. It is therefore proposed to look at the Austin unit first as the author is obviously more familiar with this and, as with the review of the automobile units, the aims and high lights of other companies will be observed.

All the industrial units in this section are single-shaft machines.

Austin Motor Company

The 250-bhp unit (14) is a single-shaft machine consisting of a single-stage centrifugal compressor driven by a two-stage axial-flow turbine (Fig. 18). The design philosophy has been to make a unit that is robust without being unduly heavy. For this reason aluminium castings have been used wherever temperature will permit; this includes the gearbox and compressor castings. A pressure ratio of 3·5/1 was chosen as the maximum desirable for a single-stage compressor from the standpoint of stress. A two-stage turbine was selected from efficiency considerations. Constant-section blading has been used as the deviation from free-vortex blading is not great with the hub/tip ratio employed in this design. It does of course materially reduce the cost of the forged and machined blading. Stresses necessitated a fir-tree root fixing if separate blades were to be used. Under the conditions of stress and temperature prevailing, Nimonic 90 is adequate for a life of 5000 hours under full-load running conditions. The turbine unit has a single combustion chamber and fuel system supplied by Joseph Lucas Ltd. One of the unique features of the fuel system is the combined sprayer igniter in which the atomized spray is ignited by a high-energy spark close to the base of the conical fuel spray (Fig. 19).

A rotor assembly with an overhung turbine rotor was adopted to allow for the easy addition of a free axial-power turbine for vehicle purposes in a later stage of development. Calculations and previous experience had shown the need to keep clearances on both the compressor and turbine to a minimum and for this reason it was decided to run the turbine entirely below its first critical speed; this necessitated a stiff rotor construction (Fig. 20). A combined journal and thrust race is used at the compressor eye and a roller bearing in front of the turbine. The whole rotor

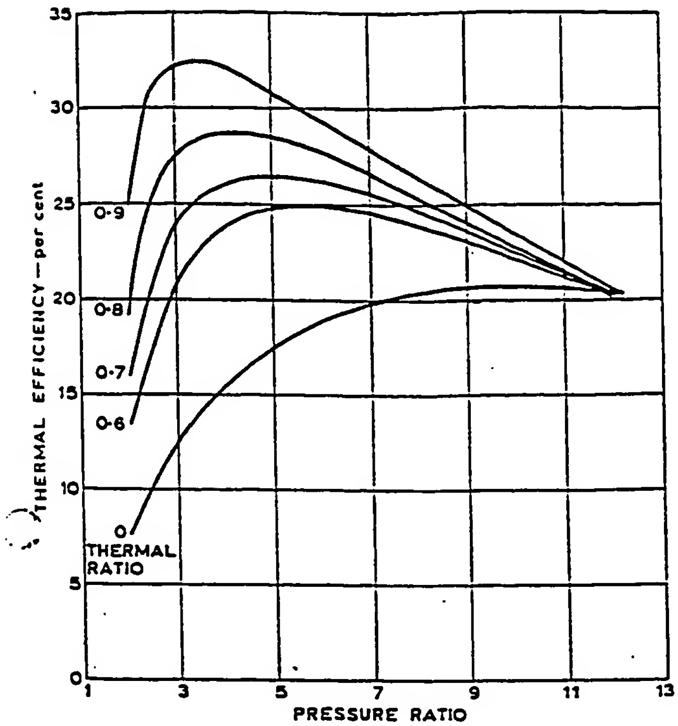


Fig. 1. Effect of pressure ratio on thermal efficiency

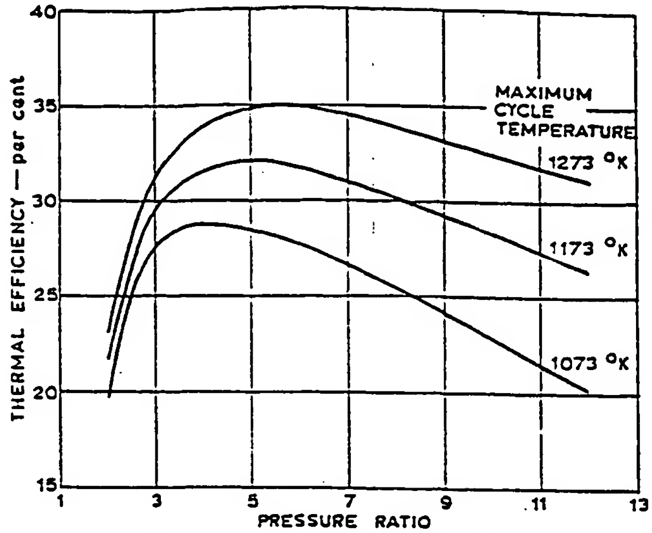


Fig. 3. Effect of turbine inlet temperature on thermal efficiency

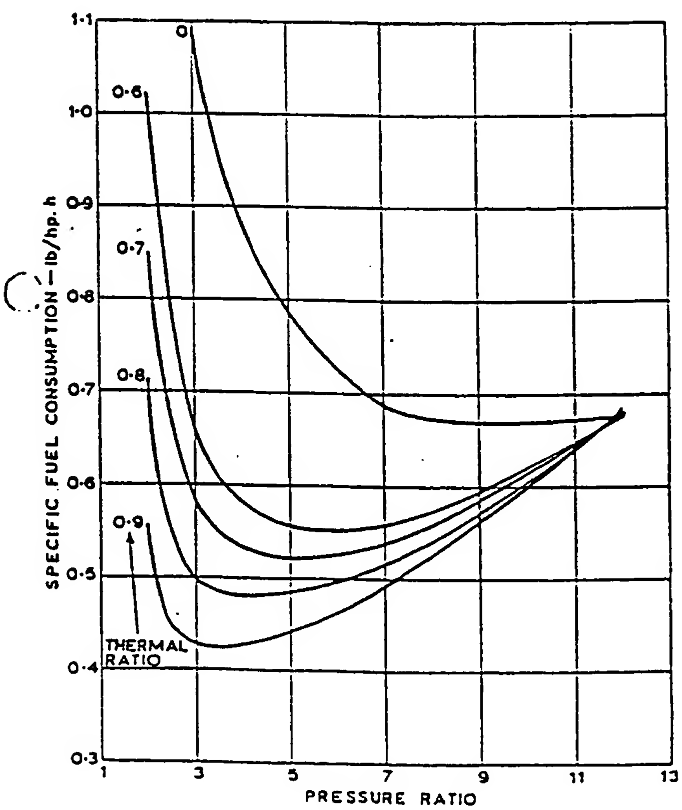


Fig. 2. Effect of pressure ratio on fuel consumption

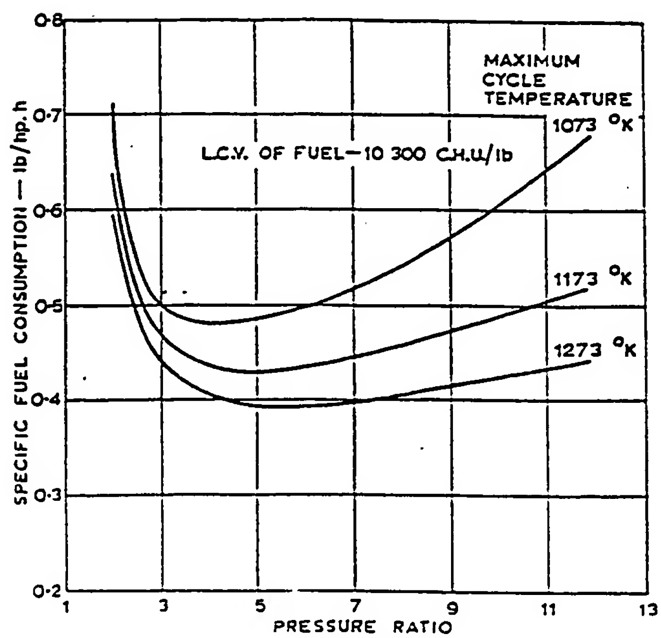
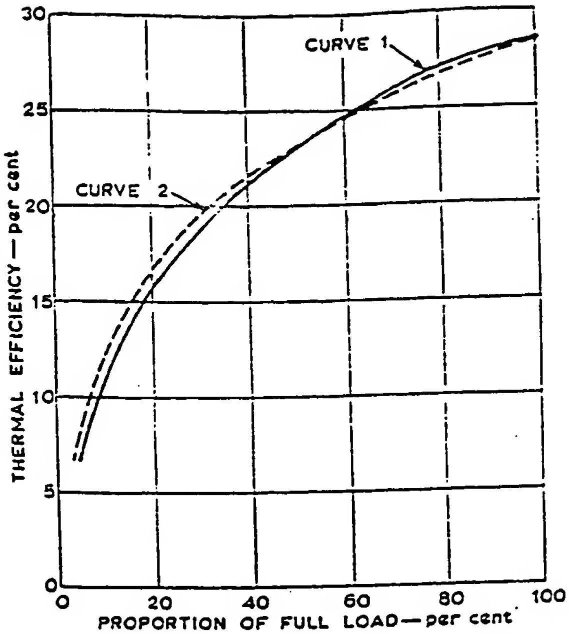


Fig. 4. Effect of turbine inlet temperature on fuel consumption



Curve 1 ————, engine with heat exchanger; thermal ratio 0·8, design point pressure ratio, 3·5; design point T_{max} 1073°K (1931°R); design point compressor efficiency 80 per cent; turbine efficiency 85 per cent.

Curve 2 — — — —, engine without heat exchanger; design point pressure ratio, 14·0; design point T_{max} 1433°K (2579°R); design point compressor efficiency 76 per cent; turbine efficiency 85 per cent.

Fig. 5. Theoretical comparison of heat-exchanger unit with a high temperature no-heat-exchanger unit

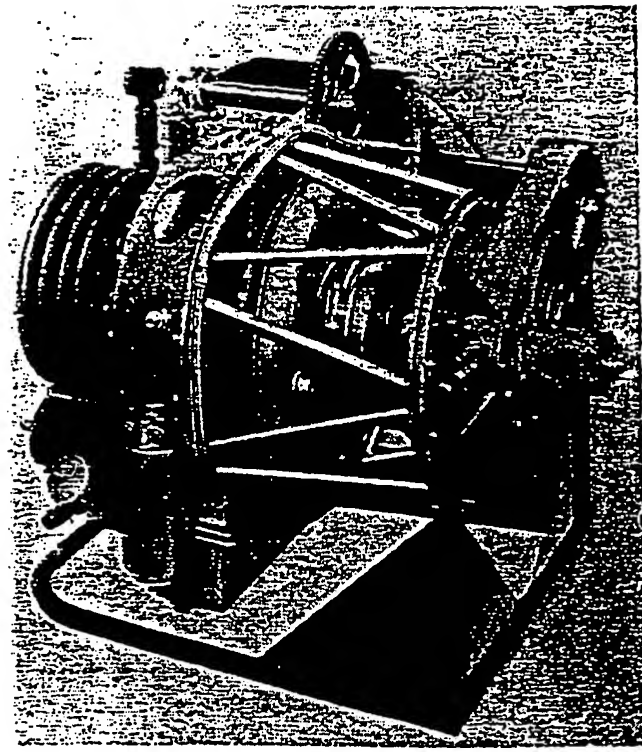


Fig. 7. Austin 30 hp two-shaft turbine

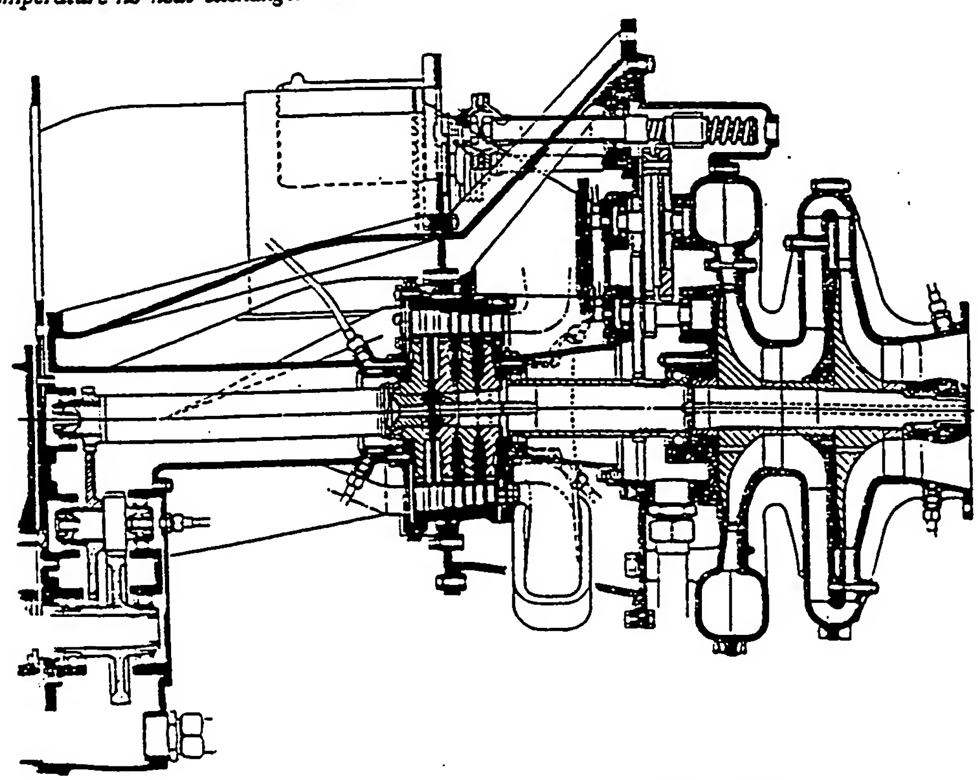


Fig. 6. Cross-section of Austin 120 hp vehicle turbine

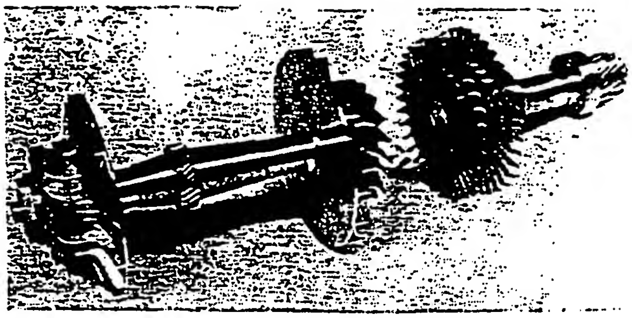


Fig. 8. Rotors of Austin 30 hp two-shaft turbine

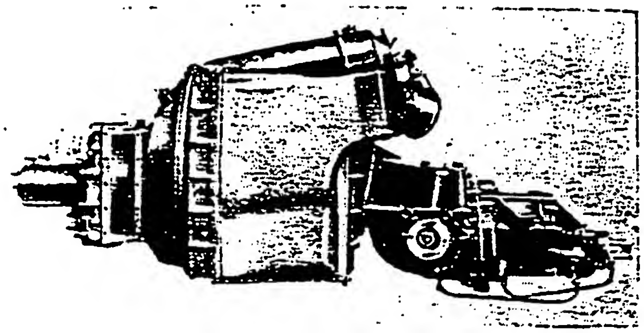


Fig. 9. Rover 2S/140 turbine engine

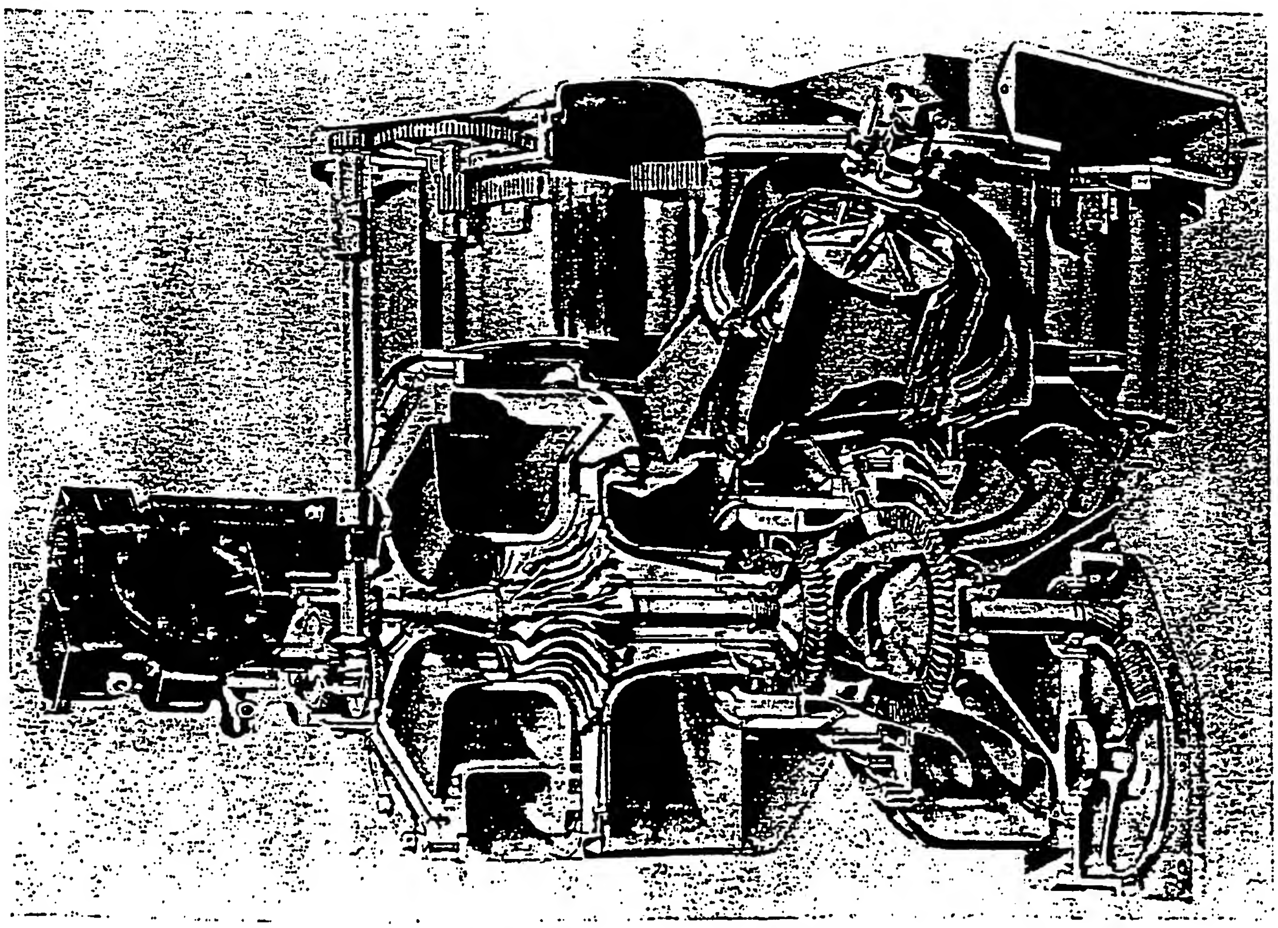


Fig. 10. Chrysler CR2A turbine unit

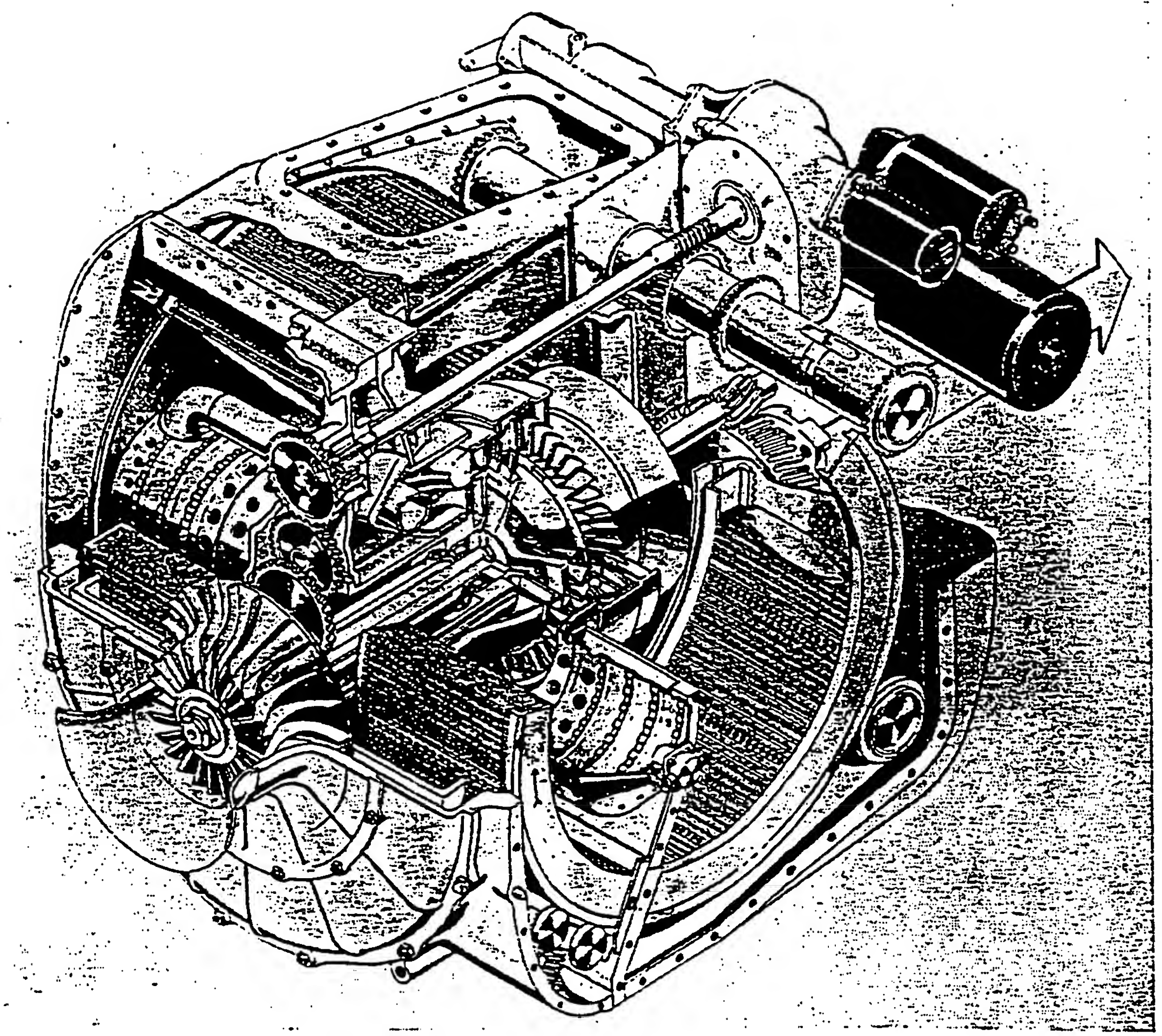


Fig. 13. General Motors GT305 Whirlfire engine

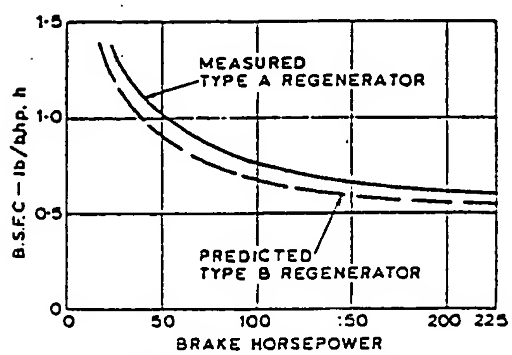


Fig. 14. General Motors GT305 fuel consumption

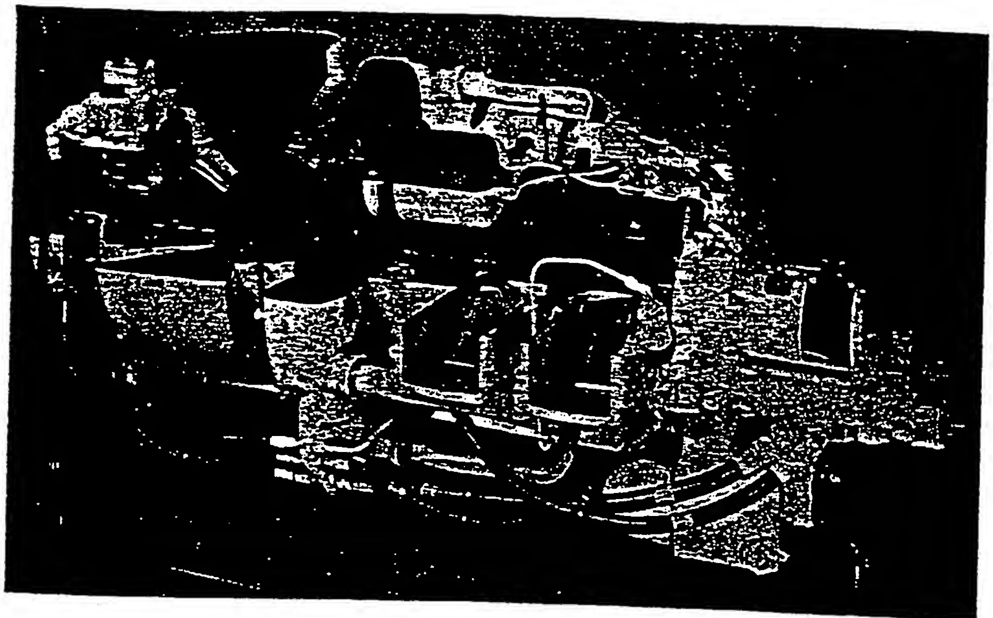


Fig. 15. Boeing 520 two-shaft gas turbine engine

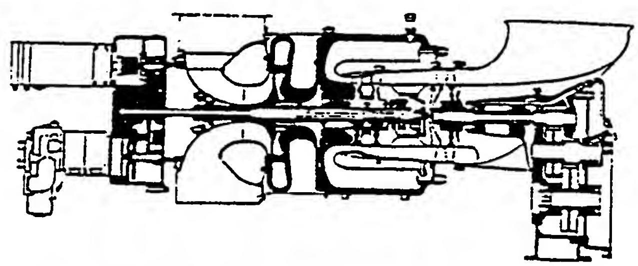


Fig. 16. AiResearch Model 331 gas turbine (without recuperator)

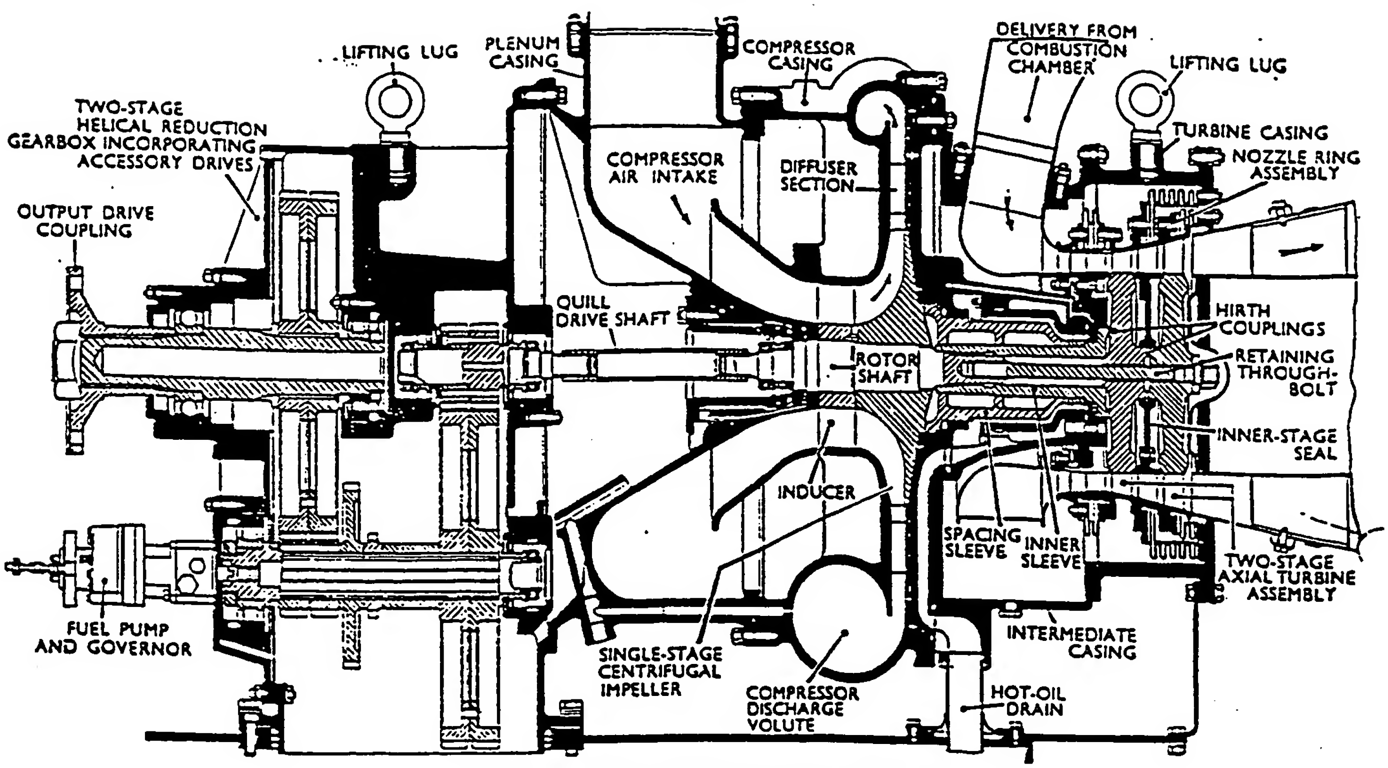


Fig. 18a. Austin 250 industrial gas turbine

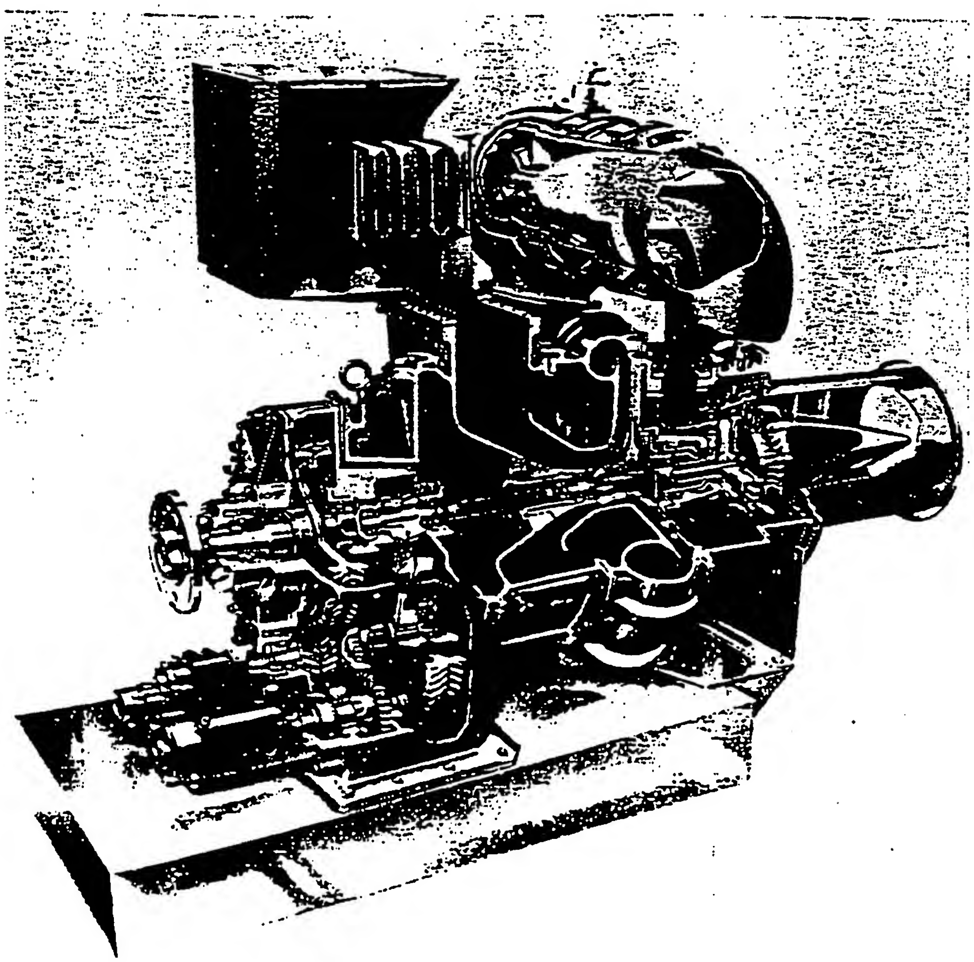


Fig. 18b. Austin 250 industrial gas turbine

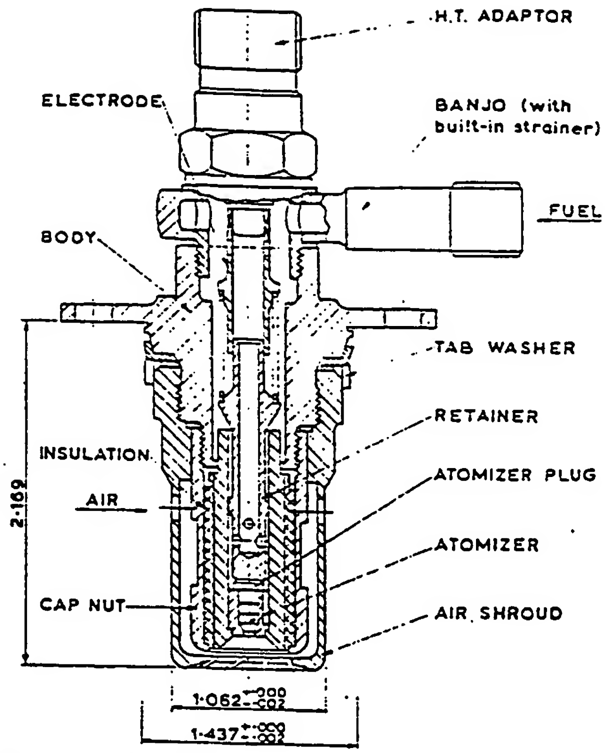


Fig. 19. Austin 250 industrial gas turbine, Lucas sprayer igniter

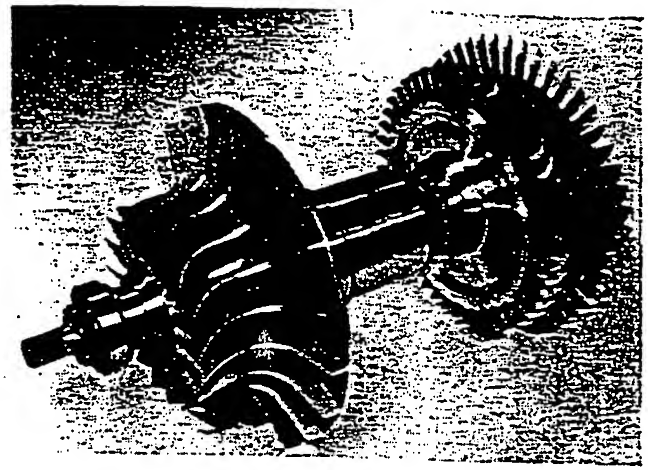


Fig. 20. Austin 250 industrial gas turbine, rotor

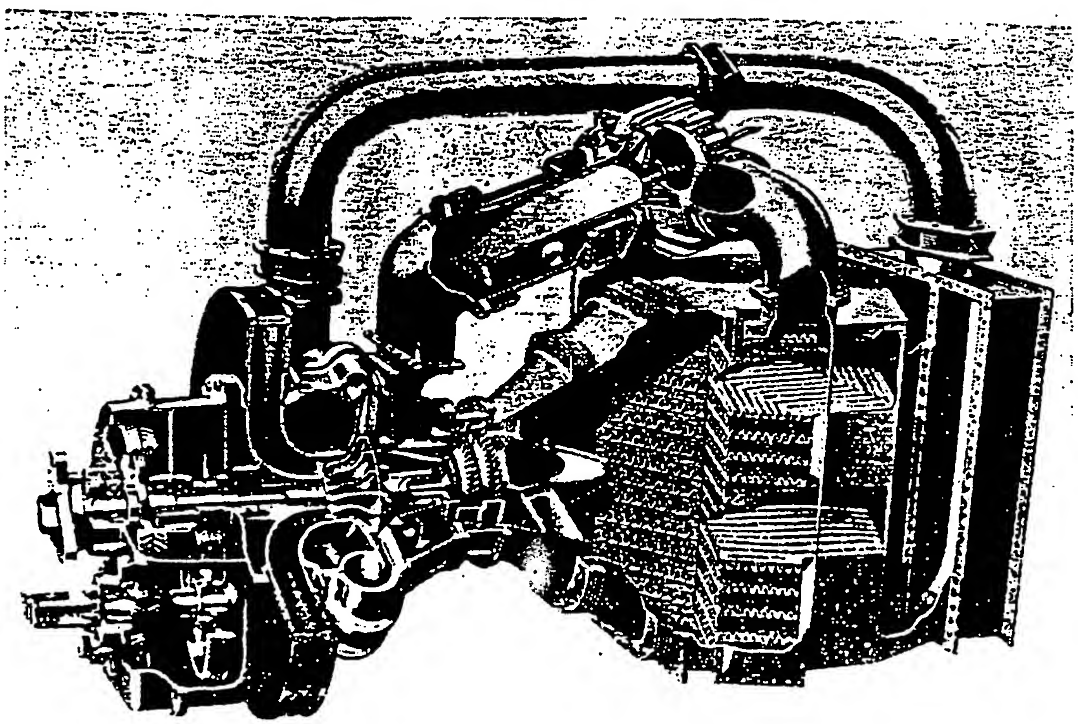


Fig. 21. Austin 250 industrial gas turbine, with heat exchanger

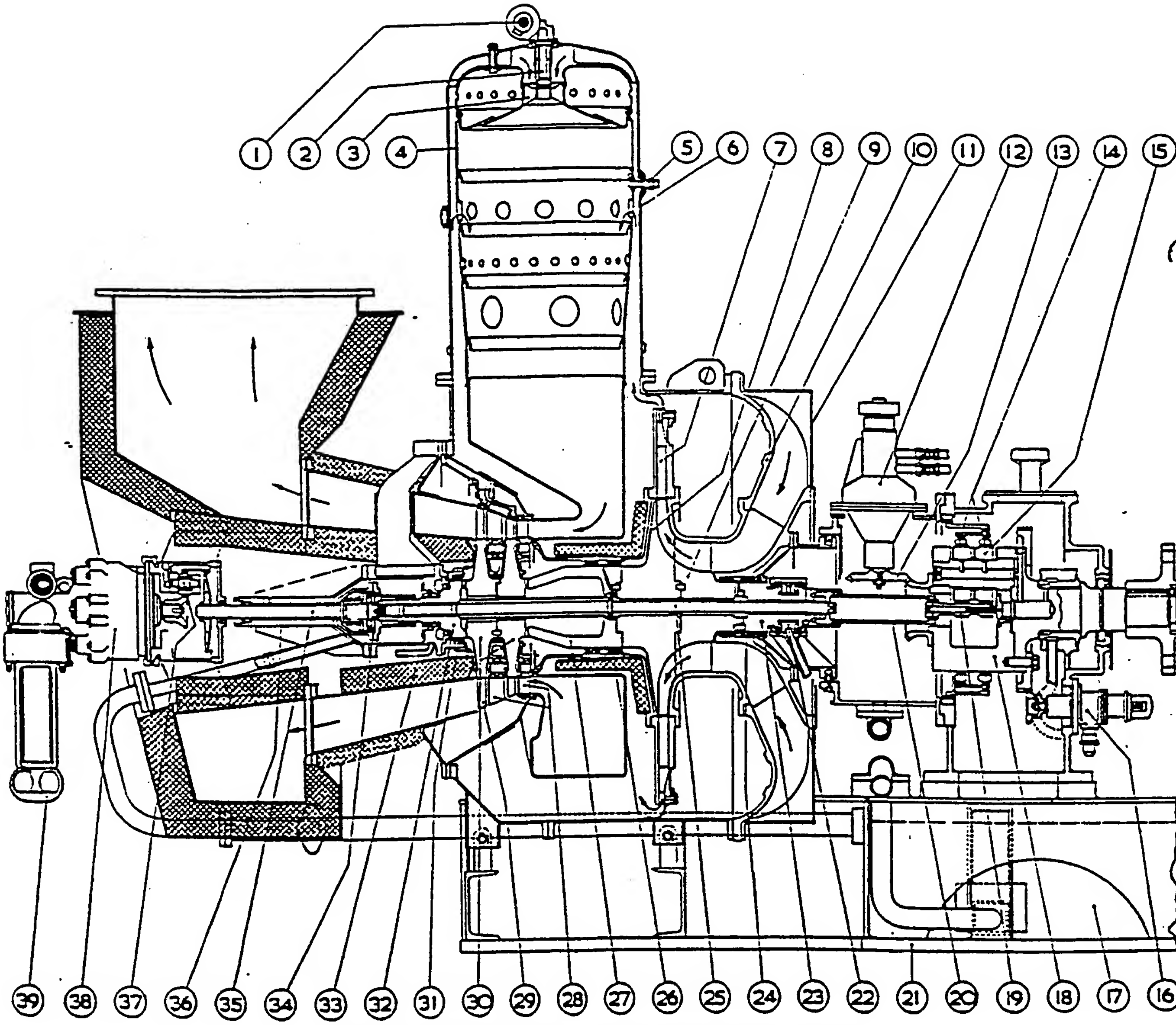


Fig. 22. Sectional arrangement of Allen 350 kW gas turbine

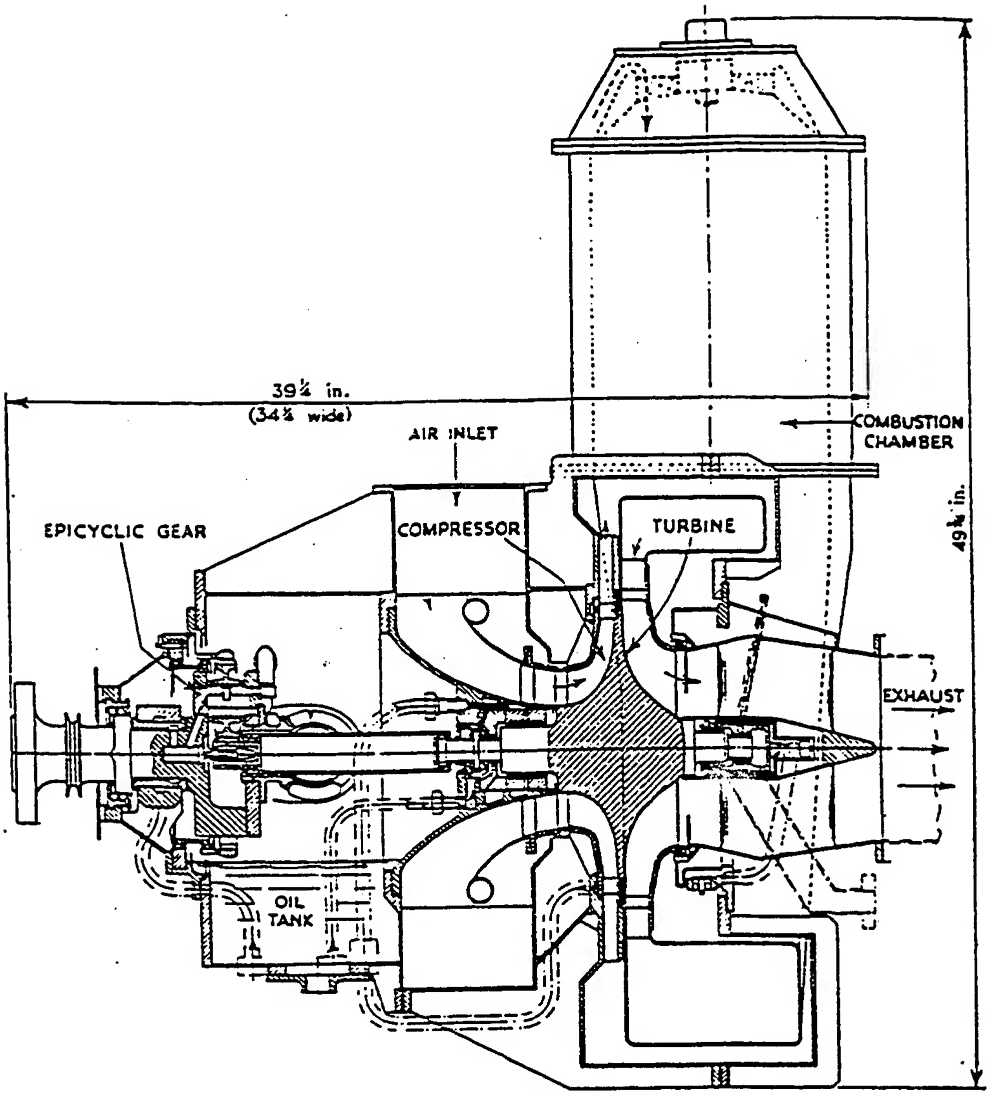


Fig. 23. Sectional arrangement of Allen 125 kW gas turbine

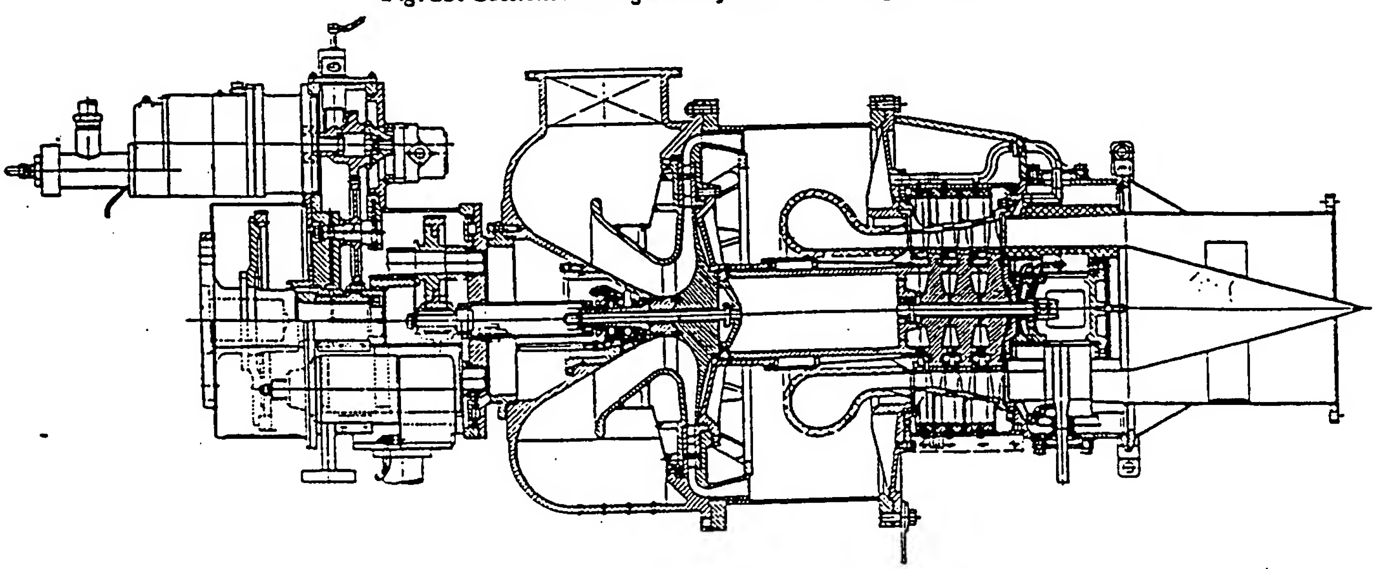


Fig. 24. Ruston and Hornsby TE 300 kW gas turbine

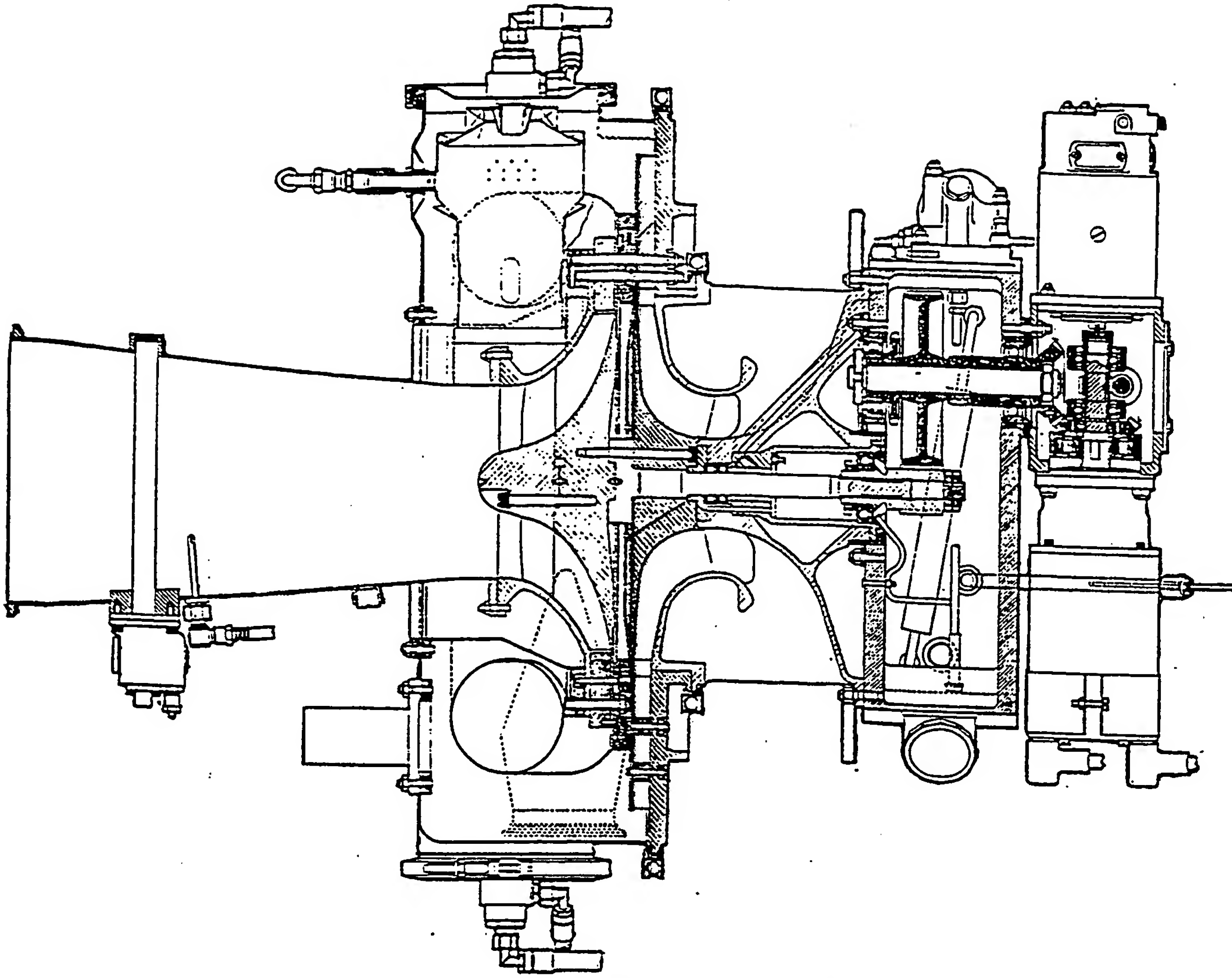


Fig. 25. Auto Diesels Ltd air-bleed gas turbine

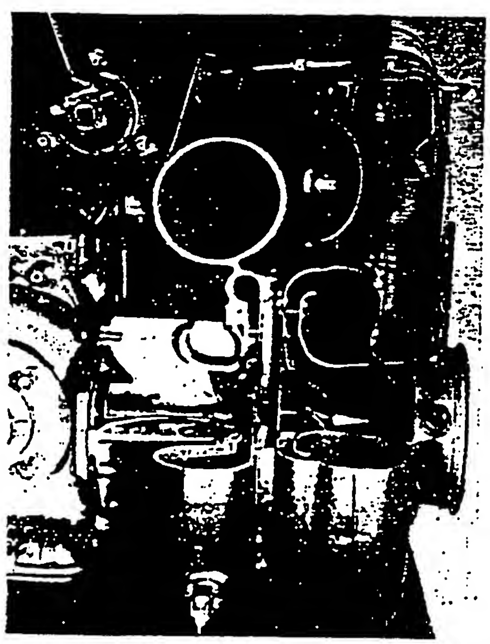


Fig. 26. Solar 'Mars' (Perkins Engines Ltd) gas turbine

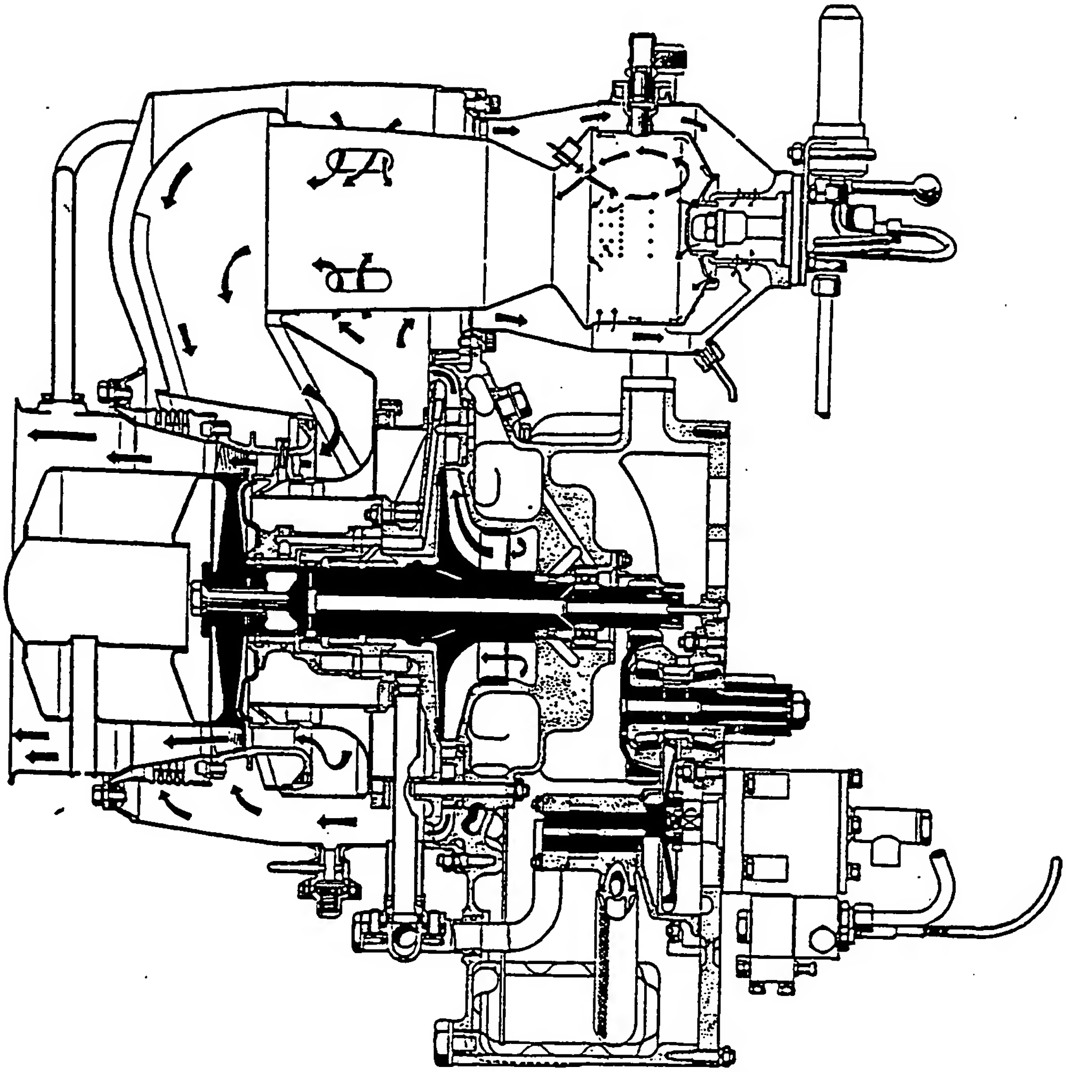
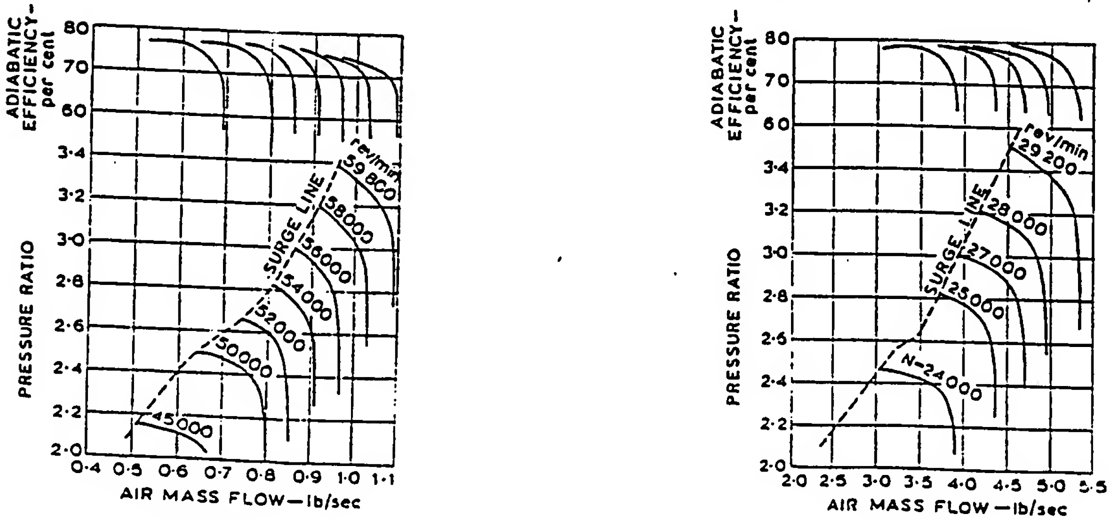


Fig. 27. Rover IS/60 industrial gas turbine



a 30 hp.

b 250 hp.

Fig. 29. Characteristics of Austin compressors

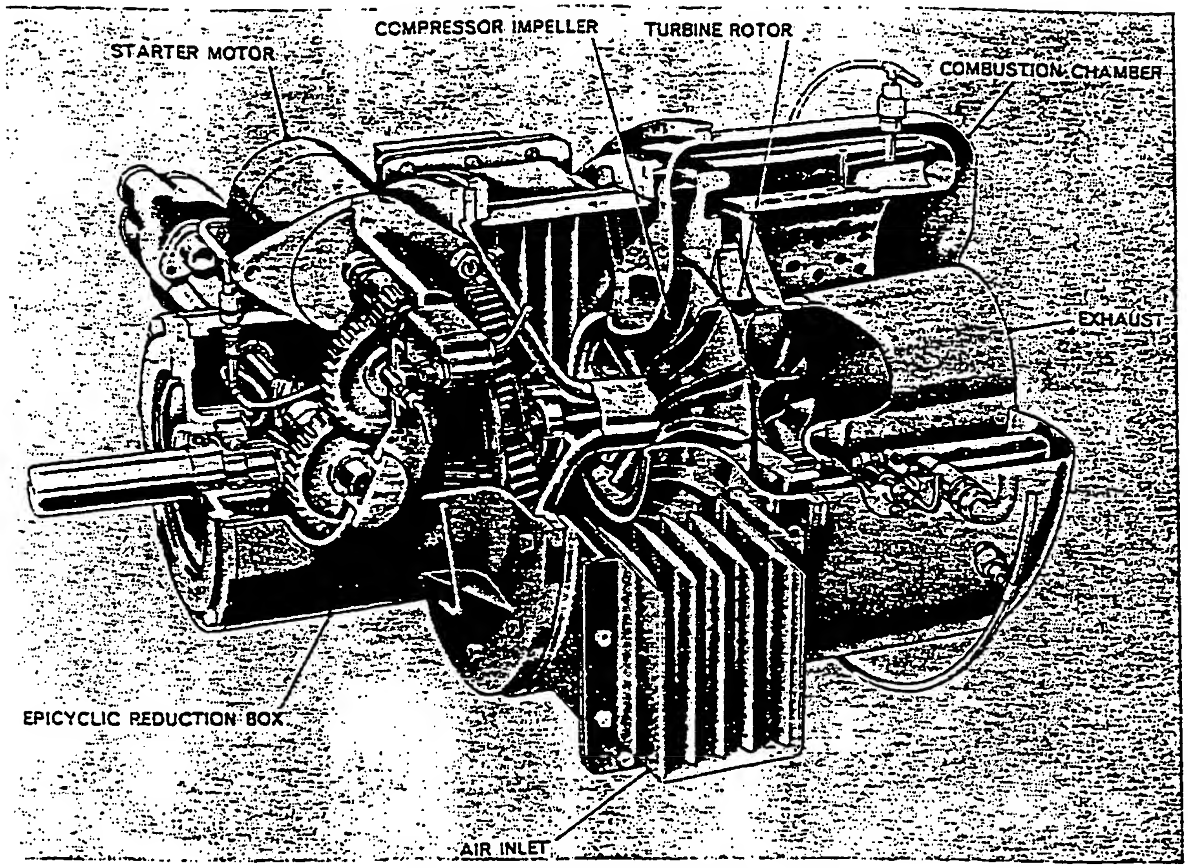


Fig. 28. Budworth 'Brill' industrial gas turbine

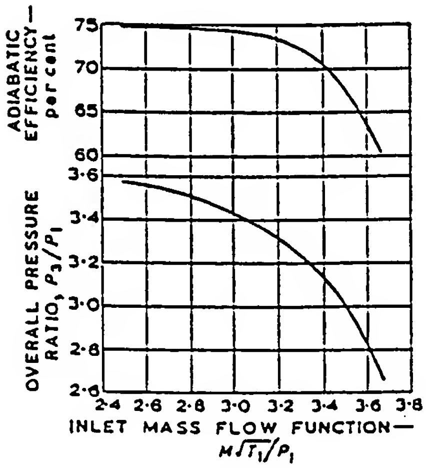


Fig. 30. Overall characteristic Austin 120-hp compressor

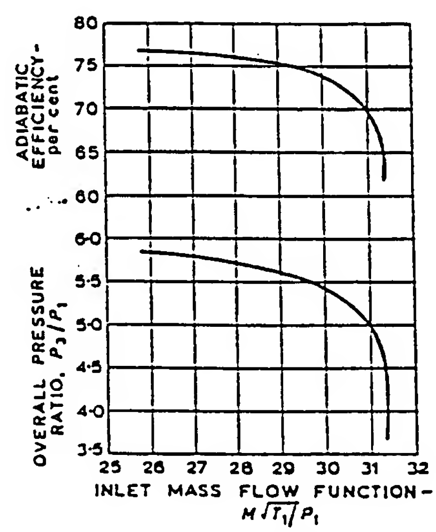


Fig. 31. Overall characteristic Rolls-Royce 'Dart' compressor

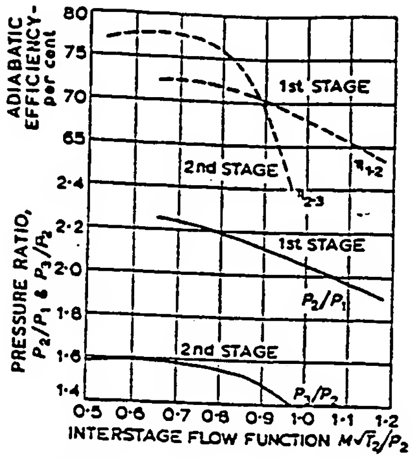


Fig. 32. Stage characteristics Austin 120-hp compressor

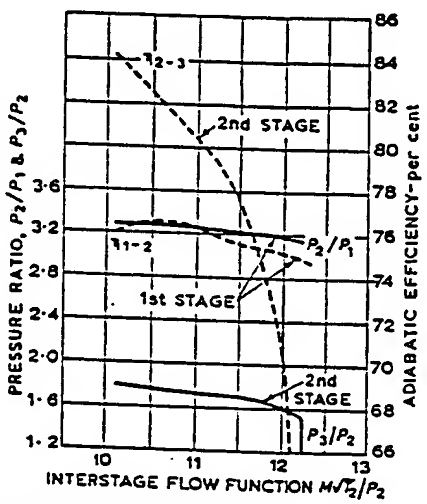


Fig. 33. Stage characteristics Rolls-Royce 'Dart' compressor

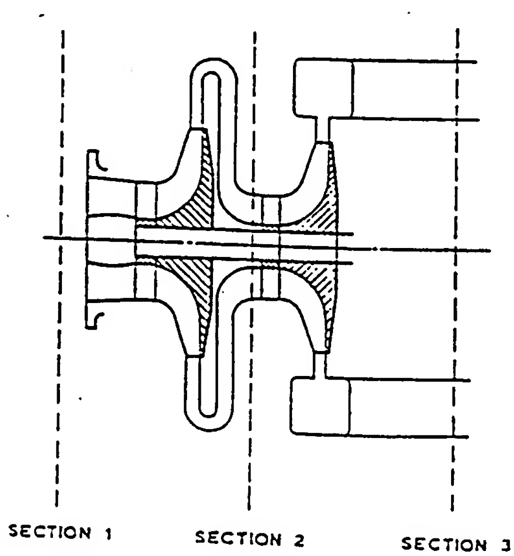


Fig. 34. Location of measuring sections

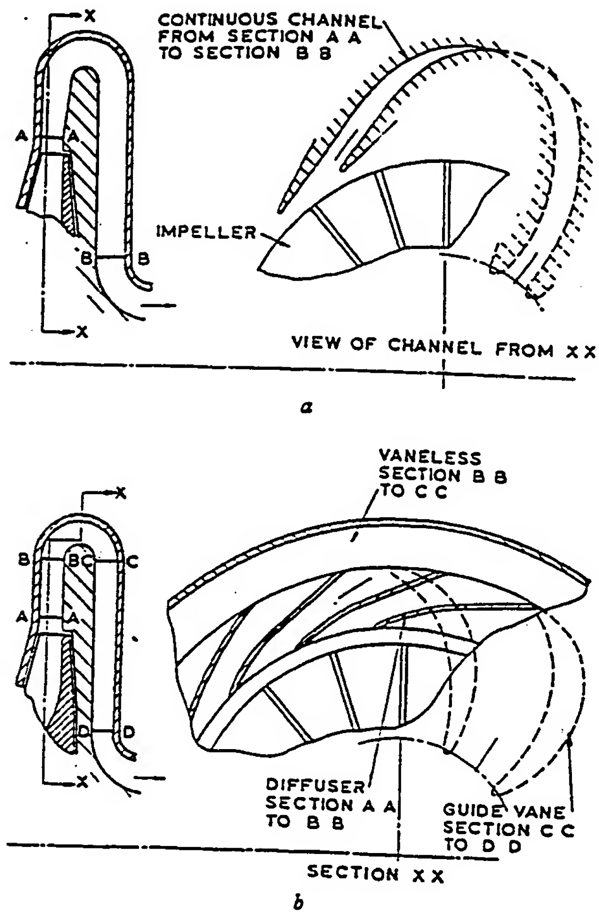


Fig. 35. Diagrammatic arrangement of (Mk. 1) aerofoil vane diffuser and (Mk. 18) channel diffuser

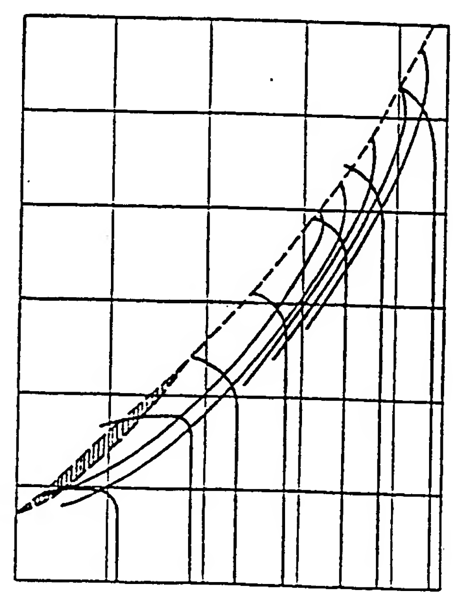


Fig. 36. Compressor characteristic Boeing T.60 compressor

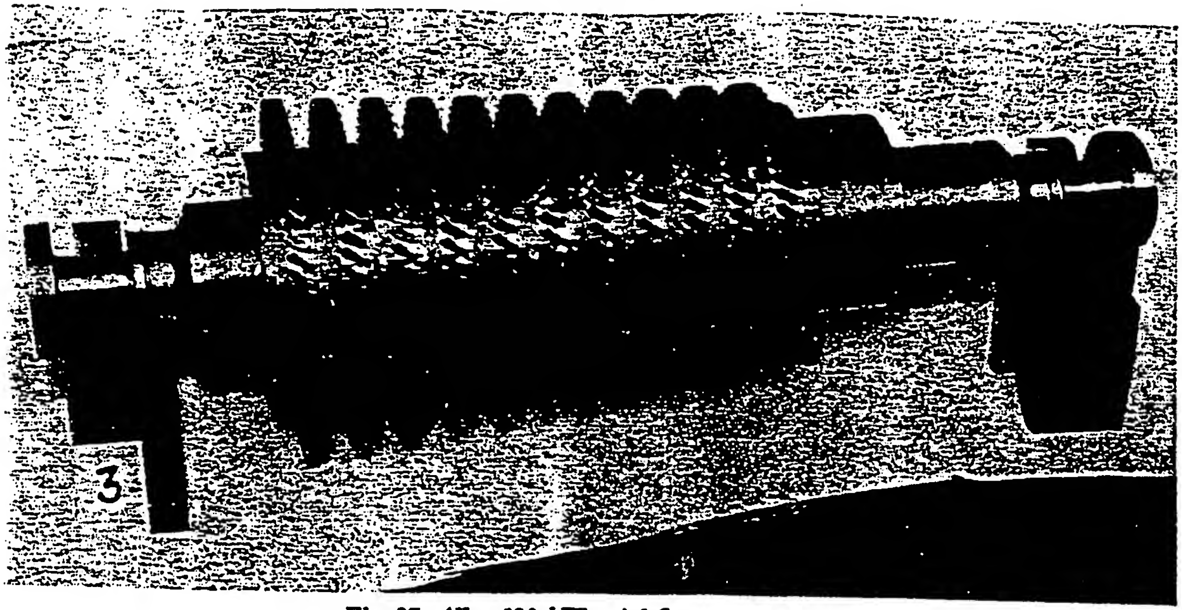


Fig. 37. Allen 500-kW axial flow compressor rotor

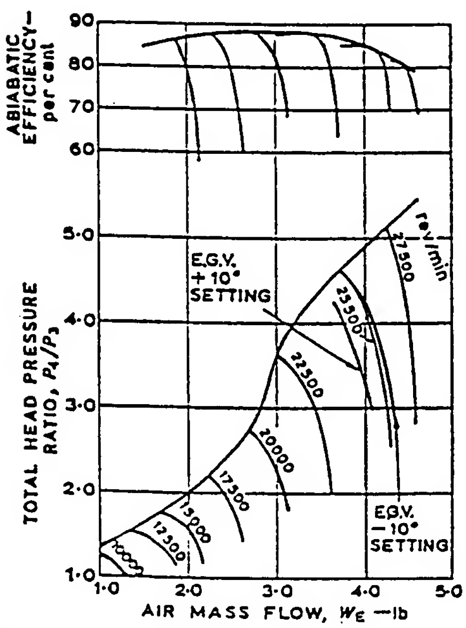


Fig. 38. Compressor characteristic of Allen axial flow compressor

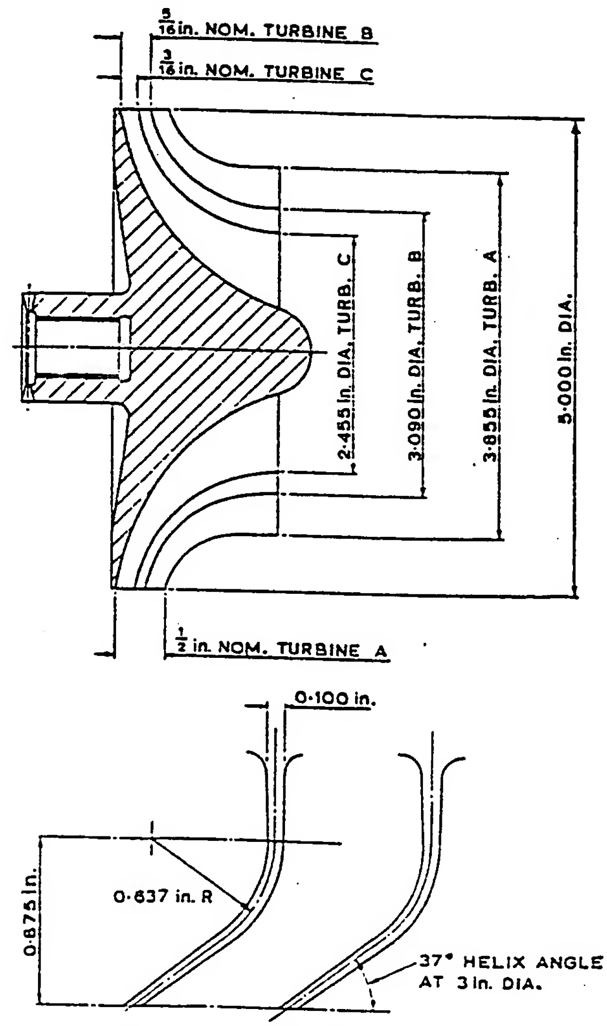


Fig. 39. Radial inward-flow turbine rotors A, B and C, Ricardo

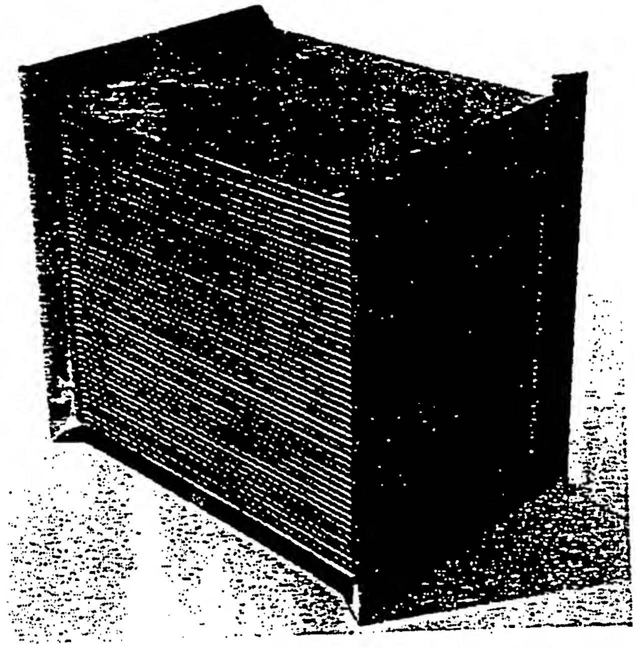


Fig. 40. Austin cross-flow heat exchanger block

Fig. 41. Austin heat exchanger rig

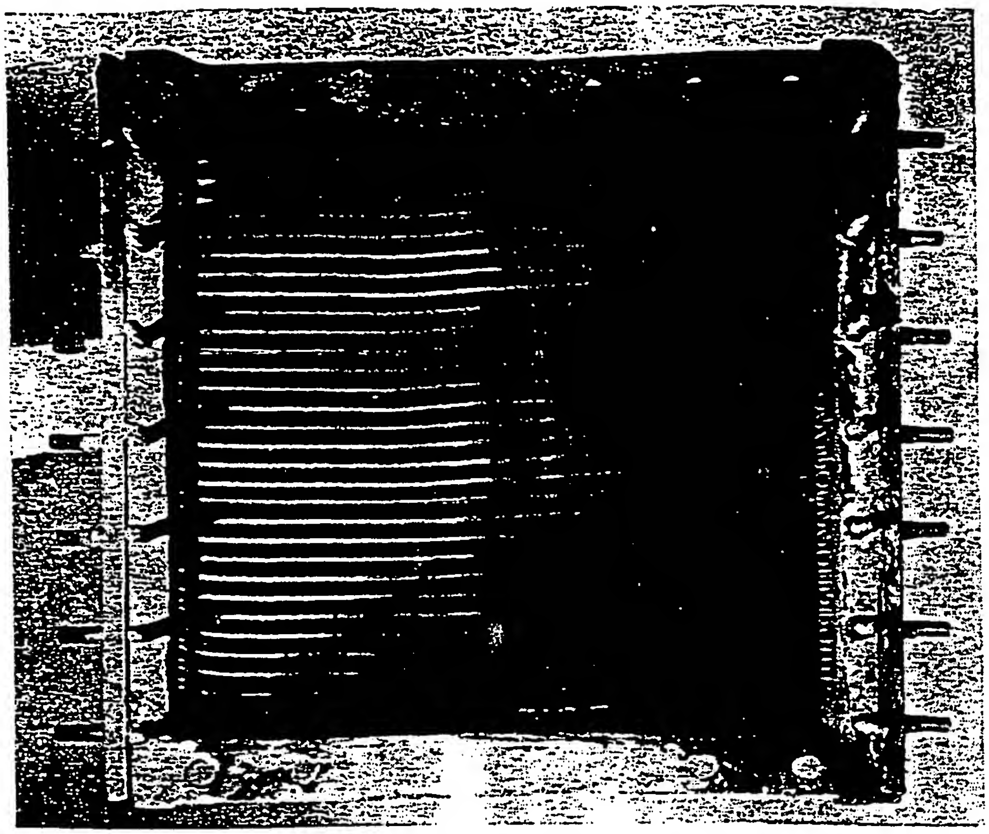


Fig. 42. Gas inlet face of Austin 30-hp cross-flow heat exchanger block after endurance test

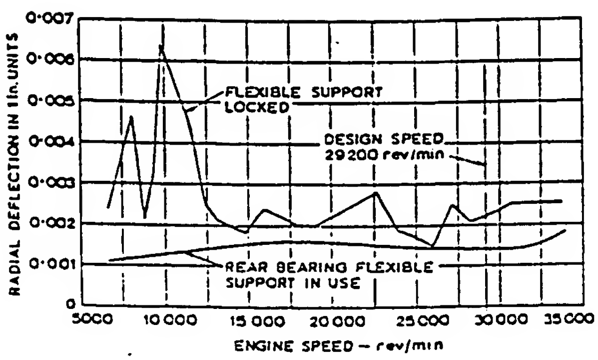


Fig. 43. Austin 250-hp rotor vibration characteristic

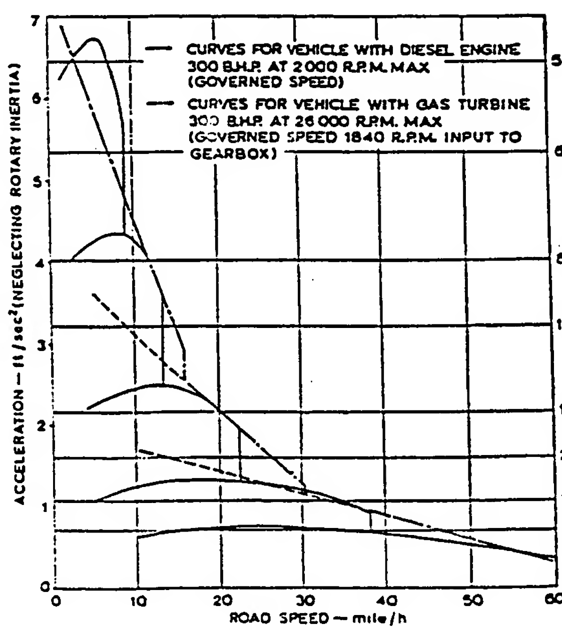


Fig. 44. Calculated performance curve for 24-ton truck powered with 300-hp gas turbine and 300-hp diesel engine

assembly is held together by one central bolt which is carefully prestressed.

The gearbox consists of two sets of double-helical gears; the ratio of the second pair of gears may be varied to suit different applications. All auxiliaries are driven off the layshaft. These include a simplified piston-type fuel pump combined with a centrifugal governor developed by Joseph Lucas Ltd, a gear-type lubricating pump, and an electric starter which drives through a sprag-type free wheel.

To convert the unit to the contra-flow heat-exchanger version requires new ducting and a double-entry combustion chamber but all other changes are minor (Fig. 21).

W. H. Allen

The Allen 350 kW (15), designed probably a little before the Austin turbine, though double the power output, is perhaps the unit bearing the closest resemblance (Fig. 22). Both use the same major components but, while Austin use an elbow combustion chamber the Allen is a reverse-flow type. This is probably more compact though it requires a little more head room. It was not used in the Austin design as it does not lend itself to the addition of a heat exchanger. The rotor shaft configuration is different in that the turbine is simply mounted between end bearings. These bearings are of the plain type, both for journal and thrust. In detail, Allen's have preferred an epicyclic reduction gear to the layshaft type.

The 125 kW unit (Fig. 23), for emergency power generation, was designed for extreme simplicity and the now familiar all radial flow compressor and turbine was selected to achieve this end. An integral rotor was used so that the compressor side cooled the turbine. Calculations indicated that the heat flow to the compressor would not affect its efficiency greatly. A simply supported rotor with plain bearings was used giving a very stiff rotor. A reverse-flow combustion chamber and epicyclic reduction gearing were utilized.

Ruston and Hornsby

The successful Ruston 'TE' 300-kW unit (16) differs mainly from the Allen in the addition of an extra turbine stage. This is logical as the Ruston unit has a higher pressure ratio, namely 3·5/1 compared with 2·5/1. In addition, this allows a lower rotational speed though similar to the Allen in basic layout. It will be seen that it differs considerably in detailed design (Fig. 24).

The main shell is built up of four substantially annular castings spigoted and flange-bolted together, the gearbox, the compressor, the turbine inlet casing and the turbine casing. The engine has been designed for a long life of 25 000 h which for the comparatively high gas inlet temperature of 800°C (1472°F) would be a good achievement. Rotor blades are machined for free-vortex flow and fixed to the discs with fir-tree roots. Stator blades, which are of constant section, are assembled in eight segments to cater for expansion problems.

The rotor is supported at its ends on plain bearings and the centrifugal impeller and the turbine discs are separated by a tube of large diameter giving a very stiff rotor, the bearings are elastically supported.

Auto Diesels Ltd

Manufacturing under licence an engine originally developed by the Standard Motor Co., Auto Diesels Ltd have developed an air-bleed unit used mainly for starting large aircraft jet engines (Fig. 25). The illustration shows a compact engine with radial compressor and turbine of back-to-back construction and overhanging rotor. Two combustion chambers are utilized, presumably to keep the outside dimensions as small as possible. For an industrial unit it has a comparatively high turbine inlet temperature of 850°C.

Solar

In the small industrial turbine, Solar and AiResearch were the first in the field. The Solar 'Mars' unit of 50 hp (18), which is manufactured in Great Britain by Perkins, broke new ground by utilizing the back-to-back construction for the turbine and compressor.

This is a good layout for a light-weight construction as it gives the shortest possible configuration. Turbine and compressor rotors are separate and overhung (Fig. 26) and held on a slender shaft. Clearly, the system runs above the first critical speed and, although this is a disadvantage in view of having to accept a deflection while passing through the critical speed, it does allow small bearings of lower friction drag. Sealing at a small radius between the rotors is accomplished by a diaphragm. It will be noted from Table 1 that this unit is among the lightest and is conservatively rated for emergency duties with reliability. The ducting layout contributes considerably to the light weight and the connection between compressor and turbine is accomplished largely by the elbow combustion chamber.

AiResearch

AiResearch were the first to enter the field of industrial gas turbines and they have manufactured several thousand air-bleed units for aircraft air-conditioning and pressurization. They have been pioneers of the inward-flow radial gas turbine and have done much to develop and popularize this type of turbine for small units.

Their engines range in size from 30 to 850 hp (19) and with the exception of the 331 series described above are single-shaft machines. Most units have two stages of compression. This is necessary as high-pressure air is frequently required for air-bleed purposes. The low-pressure compressor is a symmetrical double-entry centrifugal unit, the output of which, now at higher density, is fed to a single-entry radial compressor, both impellers being driven on the same shaft by an inward-flow radial turbine. In many cases engines giving bleed-air and shaft power are available. One of their latest and most interesting engines is the GTP 30.1, which gives 30 hp with a weight of only 40 lb, and is undoubtedly the lightest gas turbine power unit. It is a development from their turbo-supercharger, and consists of a single-stage centrifugal compressor

driven by an inward-flow radial turbine, it runs at a speed of 52 800 rev/min with a turbine inlet temperature of 795°C (1463°F).

Rover Company

The Rover Company has developed two robust industrial units (20), the IS/60 and IS/90 of 60 and 90 bhp respectively (Fig. 27). These are practically identical but the larger unit has an increased air flow. The design utilizes a centrifugal compressor driven by an axial turbine. In detail there are many interesting features. The 17-vane impeller is made, as is common practice, in two pieces but the rotating inlet guide vanes are shrouded. The turbine is a Nimonic forging with the blades machined integrally with the disc, and as these conform to a free-vortex design it represents quite a feat in machining. The ball journal and thrust race is of standard high-speed practice but the diameter of the roller bearing has been reduced by the expedient of a groove in the turbine shaft to form the inner race while the outer race forms part of a flexible bearing mounting. The combustion chamber is of the reverse-flow type.

D. Budworth

David Budworth, being a small company, was very daring when it embarked in 1950 on the development of small gas turbines. Nevertheless, the company has succeeded in marketing a successful machine in the 'Brill' (21). The original philosophy of the design was extreme simplicity, the elimination of electrical accessories, and the keeping to a very minimum of other accessories. It is an all-radial-flow single-shaft machine (Fig. 28) with an annular combustion chamber. In order to extend its range of application the Mark II has been developed, with increased horsepower (60–90 hp) and electrical accessories have been made available. The annular combustion chamber differs from the Turbomeca design in that fuel is fed from three static upstream-injection burners and pilot atomizer. Starting can be by hand cranking or an electric motor if preferred.

B.M.W. and Deutz

German firms entering into the turbine field somewhat later than other industrial countries have rightly profited by the latter's experience. B.M.W. (22) use a similar all-radial-flow design in a 50 bhp unit to that pioneered by Solar and have obtained even more compactness by utilizing an annular combustion chamber of the same principle as the Turbomeca. Their unit is thus similar in construction to the Budworth design.

Klockner–Humboldt–Deutz have under development a light unit of 80 bhp for fire-pump duty (23). This is similar in design to the B.M.W. but uses a more conventional reverse-flow combustion chamber. Less conventional but simple is the igniter arrangement, which is similar to a 'Bengal match'. This is struck and inserted into a holder where it burns for some 20 sec.

Both of the above engines have been designed for inexpensive manufacture.

ANALYSIS OF APPROACHES OF VARIOUS FIRMS

When we analyse the various approaches, although there are exceptions a definite pattern emerges for the various fields of application. In the first place almost everyone has avoided the axial compressor in this size of unit, despite the fact that its efficiencies are known to be better in the present state of the art. This is no doubt primarily due to cost. Some eight stages would be necessary for an efficient compressor of 3·5/1 pressure ratio. Coupled with this, the axial compressor has a narrow working band between surge and loss of efficiency making part-load operation difficult. Additionally, it is very much subject to loss of efficiency and stability with deposits. With a few exceptions single combustion chambers have been chosen no doubt primarily to save cost and to eliminate any possibility of one or more combustion chambers failing to light, which could have disastrous results on the engine. There is, however, divergence of view on the configuration: straight-through, reverse-flow and elbow types all have support. This, however, is mainly due to the mechanical arrangement of the design rather than combustion efficiency, which is close to 100 per cent in all cases. Where two combustion chambers have been used, namely Boeing, General Motors, Fiat, and Auto Diesels, this is presumably to give a more compact engine with the configuration employed. The Ford unit has two combustion chambers, one being necessary for reheat.

These observations are common to both industrial and vehicle units but to discover more detailed trends it is necessary to consider the two fields separately because of the influence of the different objectives.

Automobile type turbines

All automobile type turbines are two-shaft units; some of them have been developed for aircraft purposes but would be suitable for vehicles and, in fact, in many cases have been used experimentally in this manner. With the exception of the Ford, Fiat, AiResearch and Austin 120 hp a single-stage compressor (ratio 3 to 4/1) has been used, requiring a heat exchanger of high thermal ratio. In the early days it was not generally thought that a heat exchanger of such a large thermal ratio would be a practical proposition for the small space available in a car. With a thermal ratio of 0·7 a pressure ratio of 6/1 could be advantageous, generally requiring 2 stages of compression (Fig. 1). As no small heat-exchanger turbine unit has yet been put on the market the question of pressure ratio still cannot be said to be resolved. However (see below) the combining of two centrifugal compressors is only performed with the sacrifice of efficiency, so except for no-heat exchanger units where a high pressure ratio and temperature are essential for efficiency it seems unlikely that two stages will be popular. A partial move in the direction of high pressures has, however, been made by Boeing with their single-stage compressor of 6·6/1 pressure ratio. It should not, however, be ignored that with two stages the discs are not stressed to the same

degree and there is the possibility of intercooling as used by Ford.

Axial turbines seem to be the more popular because of the convenience and shortness of the ducting between the compressor turbine and the power turbine but, as seen above, there are notable exceptions, Rover, Austin 30 hp, Boeing 520 and Ford (high-pressure section). Power turbines are all axial except for a unit in early stages of development by AiResearch. For specific weight the prize would seem to go to Boeing and Turbomeca with a figure of about 0·7 lb/bhp, not surprisingly as they are of no-heat-exchanger type and designed specifically for aircraft use; nevertheless, they are both cleverly designed units. With axial turbines two or more stages can improve efficiency, as with axial units the losses increase with increase of gas deflection or blade camber. In addition, staging allows a lower stressed design making higher temperatures or simpler blade fixings possible. In this field all are heat-exchanger units except Boeing, Blackburn (Turbomeca), Fiat, and Daimler-Benz, although it is reported that the latter have a heat exchanger under development. Turbomeca and Boeing units are primarily designed for the air, namely small turbo-prop engines and helicopter units, and heat exchangers are usually too heavy for such applications. This is, however, the component which exhibits the biggest divergence in design. Recuperative units are used by Austin, Rover and AiResearch but all different in one way or another. Rover use a primary-surface type of contra-flow design, while Austin, Ford and AiResearch all use the secondary surface matrix type, but while the Austin is a cross-flow, the AiResearch is a cross-flow but with two passes on the air-side and the Ford a contra-flow unit. The Austin industrial 250 bhp unit will be a contra-flow type but in construction by no means identical with Ford. It is seen, therefore, that no finality on the best heat-exchanger design has been reached.

Industrial units

Almost universally, for reasons already noted, manufacturers have chosen the field in which the turbine excels, and for which fuel consumption is relatively unimportant, namely emergency power, pumping, and purposes in which the exhaust heat may be utilized. In addition, though not strictly industrial, a further application has proved popular, namely to provide the power unit for driving aircraft auxiliaries to save having to tap the main engines or to run the latter when the aircraft is on the ground. In these cases light weight is the important factor.

Most firms have used single-stage units for both turbine and compressor. The exceptions are, the Allen 350 kW, the Ruston TE, and the Austin 250 bhp, all these being 250 bhp or over. Here the high pressure ratios and mass flow make two or three stage turbine units justifiable on account of efficiency. Below 250 bhp the choice, with the exception of the Rover industrial units, has been for an inward-flow radial turbine; the Rover unit has blades integral with the disc. The main reason for this choice was a stress one. It is very difficult to design a root fixing for a small gas-turbine wheel that will stand the high peripheral speed and, consequently, stress, necessary to obtain good efficiency in one stage. The second factor is an economic one. Although some of the radial rotors may, in the first instance, have been cut from the solid, many are now cast and it is undoubtedly the intention of all designers to do this ultimately. However, because of casting difficulties some have taken the safer policy of machining from the solid until the casting technique is fully developed. The unanswered question is, how high in terms of mass flow and horsepower is it possible to go with a radial turbine with profit? Many of the designers of small units have adopted the back-to-back construction following the lead of Solar. Allen's use a combined rotor for both compressor and turbine (125 kW unit); with this construction sealing must be made at a large diameter, usually that of the outside of the rotor otherwise a split seal is required. However, with the small rotors this is not as detrimental as might be supposed as the pressures in the compressor and turbine at this point are of a similar value. Nevertheless, with most units separate rotors are preferred and sealing is usually accomplished at a small diameter.

Maximum gas temperatures are about 800°C (1472°F) and fuel consumption varies over a large range from 0·8 to 1·5. This range may seem surprising but the high value of fuel consumption is not really important for emergency operations and some designers prefer the reliability that goes with lower temperatures.

PROBLEMS OF SMALL GAS TURBINES

Introduction

As will be appreciated from the last section most companies have now had opportunity to weigh up the advantages and disadvantages of the turbine as a power unit. These are now well known but are listed below to indicate the most fruitful directions of research and development.

Advantages

(1) High specific power.
(2) Built-in torque converter (for 2 shaft machines).
(3) No rubbing parts like pistons.
(4) No reciprocating parts.
(5) Easy starting.
(6) Any distillate fuel.
(7) Low lubricating-oil consumption.
(8) No cooling water.
(9) Non-toxic exhaust.
(10) Little maintenance.

Problems

(1) Heavy fuel consumption.
(2) High peripheral speeds.
(3) Acceleration of gas generator.
(4) Noise.
(5) Absence of engine braking.

As observed earlier the problem of heavy fuel consumption of the simple gas turbine is principally tackled by the addition of a heat exchanger. Nevertheless, improvement in the efficiencies of the turbine and compressor materially

assists in the increase of thermal efficiency and can thereby allow a reduction in size of the heat exchanger with consequent saving in cost, volume, and weight. It is proposed shortly to examine some of the developments made in these units, principally by the author's firm, under these headings.

The main problem of high peripheral speeds is one of materials to stand up to the high stress imposed, and in the case of the turbine also under conditions of high temperature. But apart from stating that improvement and reduction of cost of high-temperature materials are perhaps the most important of all developments for progress in the gas-turbine field, it is not proposed to treat this subject, as it would require a paper in itself to deal with it adequately.

Another problem associated with high speed is shaft whirl, which will be discussed in some detail. Finally, a few observations will be made on the problems of aircraft noise and engine braking.

Compressors

One of the main unknown factors in the application of aircraft turbines to small turbines was the scale effect. To give some indication of this, comparisons are made between the compressors of the Austin 30 hp turbine and the Austin 250 hp turbine for single-stage compressors and between the Austin 120 unit and the Rolls-Royce 'Dart' for two-stage compressors. There is always difficulty in making comparisons unless the design principles are identical; certain factors frequently make this impossible but the designs considered have a similar basis, as the author's company received some assistance from the background experience of Rolls-Royce. Each compressor has an aerofoil vane-type diffuser and pressure ratios, and hub/tip ratios are similar, and perhaps the most practical but less scientific comparison is that each has been developed to give its best performance. Fig. 29 shows the characteristics of the 30 hp and 250 hp respectively. Details of design and air velocities are given in Appendix I and the efficiencies are summarized below.

Compressor type	Maximum adiabatic efficiency, per cent	Maximum pressure ratio	Design point		
			Adiabatic efficiency, per cent	Pressure ratio	Mass flow, lb/s
30 hp	76·5	3·0	76·2	2·98	0·87
250 hp	79·5	3·51	79·0	3·47	4·7

It will be seen that the maximum efficiencies are quoted and the efficiencies at the design mass-flow. From the shape of the characteristic curve it will be observed that the maximum efficiency frequently occurs at the surge line; it is not, of course, possible to work at this point and hence operating efficiencies are a little less. From Appendix I it will be noted that design Mach numbers are highest at the impeller outlet but nowhere reach unity. This was deliberate in the design but it does not guarantee that there are no local velocities in this region approaching the acoustic velocity. It is apparent that there is an improved efficiency with the larger machine but that the difference is less than initially feared with a very small unit. This was one of the interesting outcomes of the Ministry of Supply contract (24). It is interesting to compare these efficiencies with a very large aircraft compressor such as the R.R. Derwent 5 (mass flow 61 lb/sec) which at 4/1 pressure ratio has a maximum efficiency of 82 per cent, indicating the same upward tendency with size.

If pressure ratios above 6/1 are required then it is almost essential to go to two stages or stresses become of dangerous magnitude. The characteristic of the two-stage Austin 120 compressor used on the vehicle turbine is shown in Fig. 30 for one speed only (21 000 rev/min). This is not the maximum speed of the compressor or consequently the maximum pressure ratio but a convenient speed for purposes of comparison.

The velocities quoted in Appendix I are for this condition and not the maximum that can occur. The Rolls-Royce 'Dart' characteristic is shown in Fig. 31. Results are summarized below and show the same trend as on the single-shaft units.

Compressor	1st stage		2nd stage		Overall		
	Max. efficiency, per cent	Pressure ratio	Max. efficiency, per cent	Pressure ratio	Max. efficiency, per cent	Pressure ratio	Mass flow, lb/s
Austin	72·4	2·25	77·8	1·59	74·7	3·57	2·16
Rolls-Royce	76·0	3·27	84·1	1·78	76·8	5·85	22·2

It was suspected that some losses must occur in turning the diffuser of the first-stage into the eye of the second-stage impeller and tests on each stage were conducted; the first-stage tests include the losses in turning. Figs. 32 and 33 show the results for the Austin and 'Dart' compressors respectively. Fig. 34 shows the position of the instruments. In each case it will be seen that the second-stage compressor is superior to the first stage as this does not include the turning losses. In the case of the Austin units there would appear to be some 4 per cent loss in turning. This is not surprising as the air leaves the first-stage diffuser at a Mach number of nearly 0·5. In an effort to reduce the inter-stage loss a ducted diffuser was tried in which each channel of the first-stage compressor was first diffused and then gradually turned without diffusion into the eye of the second stage in more or less helical manner. Fig. 35 shows diagrammatically the two types of construction. There was practically no improvement in maximum efficiency with this change. However, the characteristic was considerably broadened making matching without surge considerably easier.

It is not possible to draw rigid conclusions from these tests though the trend is very definite: the larger the mass flow the better the efficiency. It is not likely that there is a large Reynolds number effect as in all cases Reynolds numbers are of large magnitude but it would seem that

increased losses are due to the comparatively large surface area compared with air mass flow for small units; thus the boundary layers occupy a larger proportion of the flow area and thereby bring with them a correspondingly increased loss of energy. It is noted that this scale trend is roughly of the same order as that predicted by Balje (25).

Another interesting comparison is the remarkable single-stage Boeing compressor T.60 (26) giving 6·6 pressure ratio (Fig. 36). It will be noted that the maximum efficiency falls off above 5/1 (78 per cent) to 74 per cent at 6·6/1. Velocities in this compressor are supersonic at the impeller tip and at the entrance to the diffusers thus creating shock waves, which is a difficult flow condition to be handled efficiently. Comparisons of these figures with the two-stage compressor above indicate similar efficiencies but clearly if compression can be accomplished with one stage a more compact and less expensive design is forthcoming.

A final comparison is made with an Allen eleven-stage axial compressor (Fig. 37) for the high-pressure stage of their 500 kW unit. This was tested with atmospheric inlet conditions (27) giving a mass flow of about 4·5 lb/sec at a 5/1 pressure ratio and under these conditions falls within the range being considered. Maximum adiabatic efficiency is approximately 85 per cent; owing to the steepness of the characteristic at this high ratio, however, it may be difficult to run above 82 per cent, nevertheless, such efficiencies have never been achieved with a centrifugal compressor (Fig. 38). These results go a long way to answering the question as to what happens to axial compressors when they are made in small sizes. The answer would appear to be similar to that of the centrifugal, that is a downward trend with size, but that there is an efficiency advantage in small sizes as with large compressors in favour of the axial. It may, however, be expected that if as much research had been expended on the centrifugal unit that this differential might have been reduced.

Turbines

The question here, at least for the compressor turbine, would appear to be; radial or axial? It seems possible to design an axial turbine (Table 1) with an efficiency of about 85 per cent. Radial turbines have, however, become increasingly popular in recent years for units below 250 hp. This type of turbine was probably first used by Kapitsa as an expansion turbine for refrigeration to obtain very low temperatures at the Cambridge University Mond Laboratories. It is basically a centrifugal compressor running backwards and its attraction as a turbine is centred in the fact that it eliminates the highly stressed root fixings which are necessary for most axial turbines. If machined from the solid, however, it is expensive but as a casting becomes attractive. It also lends itself more readily to variable-area nozzles. Examples of this type of unit are shown in Table 1, where no less than nine firms have used it.

Ricardo's have done considerable investigation of this type of turbine for the Ministry of Supply (28). They utilized three sizes of rotor with identical diameter but increased the size and mass flow of the turbine by increasing the axial depth of the nozzles and the rotor blade width at entrance (Fig. 39). A summary of their results is as follows:

Turbine	Efficiency
A	90
B	86
C	78

These results were for cold tests and it is suggested in their report, from a limited number of hot tests, that there would be a loss of some $1\frac{1}{2}$ per cent under high temperature conditions. In addition, rotor/shroud clearances were from 0·009 in. to 0·012 in. for cold tests, and it was arranged by pressure balancing that there was zero rotor-tip leakage. Arrangements of this nature were desirable for test work but in practical cases there would be a leakage and it is thought that the clearance would have to be a little higher. In view of this it would seem likely that a radial turbine can be designed to give approximately 85 per cent adiabatic efficiency and there therefore seems little doubt that this type of turbine will give at least as good a result as the axial. It is therefore likely that cost and the proposed layout of the engine will determine the type to be used.

The radial turbine lends itself well to single-shaft machines but difficulty is encountered when a two-shaft unit is required. However, the difficulties are not insuperable and the Rover 2S/140 and the Austin 30 hp are examples of such a design.

Heat exchangers

Although the first heat exchanger used for the Austin automobile unit was of tubular construction and comparatively low thermal ratio, calculation showed that to obtain a reasonably higher thermal ratio in a small compass tubes would have to be so small and numerous that fabrication would be very difficult and expensive. It was decided, therefore, to use simulated triangular tubes built up from alternate layers of corrugated and flat plates into a matrix and furnace-braze the whole unit.

Perhaps the greatest difficulty that has emerged in the design and construction of heat exchangers is the problem of the 'headers' for the separation of the air and exhaust gases to and from the simulated tubes. For this reason cross-flow units were first used. These allow the ducts to come simply from both sides of a roughly cuboidal block, Fig. 40. Such a construction, however, becomes very heavy for thermal ratios in excess of about 0·7, when the contra-flow heat exchanger is preferable. However, it was found possible to use a similar form of construction for this type of unit separating exhaust from air by diverting the air and exhaust gas from the matrix in alternative directions and collecting in headers, as shown in Fig. 21. The sizes of the passages are much exaggerated in this illustration for clarity. The technique of fabrication was developed by Marston Excelsior and also by the British Motor Corporation's Radiators Division.

Heat-exchanger material was carefully considered and most alloys that contained copper were rejected owing to

possible sulphur attack from certain fuels. It was decided to endeavour to evaluate other materials in relation to two standards which were thought to represent the least expensive and best materials respectively, namely mild steel and stainless steel. It caused surprise to find how well mild steel stood up to exhaust gas, only a little below 600°C (1112°F). For this reason the major amount of development work has been performed using mild steel which, as well as being less expensive in initial costs, is easier to fabricate. The test rig is shown in Fig. 41. Cool air is supplied to the air side of the heat exchanger and after passing through is ducted to a combustion chamber where it is heated to about 600°C (1112°F). The hot gases are then ducted to the exhaust side of the heat exchanger. Experience has shown that thermal distortion was the main difficulty, and a cycle test was incorporated by the simple expedient of bypassing the hot gas to the flue and closing the duct to the gas side of the heat exchanger. These two operations were performed simultaneously and automatically by two butterfly valves controlled by a small electric motor. A cycle of 5 minutes was found to be optimum for a severe and accelerated test. Under these conditions with an inlet temperature of 600°C (1112°F) the extreme temperatures recorded by the thermocouples on the gas inlet face were 570°C (1058°F) and 200°C (392°F). Owing to the thin material these temperatures are difficult to measure accurately but they indicate the very severe differential expansion problem that has to be dealt with. The cross-flow heat exchanger stood up to several hundred hours' running but ultimately the cycle tests both distorted the matrix and pulled it away from the outer casing giving leakage between the air and gas sides (Fig. 42) (29). It was thought that an improvement could be obtained by welding the matrix to its casing instead of brazing to increase the strength of the joint. This only gave a small improvement; a design was therefore evolved to give the matrix as much freedom as possible, and at the same time to increase the length of the heat flow path between it and its casing. This was accomplished by suspending the matrix on flexible diaphragms attached to the flanges of the air inlet and outlet ducts. The whole is encased in a box and the exhaust allowed to flow into one side and out of the opposite side. Bypassing of the exhaust is prevented by a sliding seal that has only to withstand a low pressure difference. The Austin 30 hp cross-flow heat exchanger has a thermal ratio of 0·65. Two types of heat exchangers have been developed for the 250 hp unit, a contra-flow, described above, of 0·84 thermal ratio, which is now made up in eight separate blocks, and a cross-flow, Fig. 40, similar to the 30 hp unit with a thermal ratio of 0·74, made up of six blocks. Design data for these three units will be found in Appendix I.

It is difficult to express an opinion of the relative merits of recuperative and regenerative heat exchangers. The basic advantage of the regenerator is that it is self-cleaning, allowing the use of small passage sizes. It also needs less ducting than the recuperator. Its main disadvantage is the difficulty of sealing between the high-pressure inlet air and hot low-pressure high-temperature exhaust gas by means of a sliding seal or gland with small clearance; development has now reduced gland leakage to 4–5 per cent. Several practical examples have shown that both types of unit may be made satisfactorily but their lives are still somewhat unknown factors. In addition both types are somewhat costly items.

Rotor problems

Just as the reciprocating engine has its torsional vibration troubles, so the gas turbine has its bending vibration troubles, namely shaft whirl. If it is decided to run above the first critical speed and sufficient shroud clearance has been allowed to get through it, then the harmonics have to be watched but these will generally be of lower amplitude than the fundamental mode.

All the Austin units have been designed to run below the first critical speed and very considerable development has been required to obtain a rotor assembly of sufficient stiffness, which is made more difficult by the overhung turbine rotor used in this design. Although calculations were made for critical speed, because of the doubtful factor of bearing stiffness it was felt necessary to check the design by experiment and a whirl rig was made for this purpose. This was constructed by using the 250-hp turbine engine as the basis (Fig. 43). The turbine was driven from its output shaft by a variable-speed electric motor using its reduction gear backwards to obtain the necessary high speeds. The power requirement was reduced by making rotors of identical inertia but without blades. Electronic inductive-type proximity pick-ups were located halfway between the bearings and at the overhung end of the turbine rotor. With the first rotor the fundamental critical speed was only a little above the running speed of 29 000 rev/min but by increasing the section modulus of the hollow spacing shaft between the turbine and compressor the fundamental was moved well above 35 000 rev/min (Fig. 44, upper curve for overhung end). It will be noted, however, that there were still shaft deflections within the speed range. The reason for this is probably bearing support vibration. It is impossible to balance to perfection, and the rotor presents a vibrating force covering the speed range of frequency. The various parts of the engine, including the castings supporting the bearings, have their own natural frequency and if this coincides with the rotor at any speed vibration will result. In order to reduce the magnitude of the exciting force from the rotor, the bearings were given a degree of flexibility in their mountings. The very considerable improvement is shown in the lower curve of Fig. 44.

Automobile applications

The problem of accelerating the gas generator is a considerable one and as seen above many firms have spent much time and effort to mitigate this situation. The first expedient is to keep the rotating inertia to a minimum but for high-performance cars this by itself is insufficient. It is interesting to note that Rover and Chrysler have adopted somewhat different approaches to the problem, namely in

the position in which they have placed the movable blades. In the case of the Rover design the policy is to run with a high idling speed with flow reduced to a minimum and to open up the nozzles to the compressor turbine with the addition of extra fuel. This will make more power available for the power turbine to accelerate the vehicle. With the Chrysler unit the opening of the nozzles to the power turbine increases the pressure drop across the compressor turbine and thereby helps to accelerate this component and subsequently make more power available to the power turbine. Both these effects, however, are transient conditions that take place in a few seconds and without the results of tests it is not possible to say whether a complete solution has been obtained.

With regard to braking, the turbine unit has not the natural overrun braking of the reciprocating engine. However, with modern disc brakes this is not so important with cars, though heavy trucks would certainly need some assistance in the descent of steep hills. Possible methods are firstly, the expedient of Chrysler already noted, and secondly, a method demonstrated by Boeing, namely by putting the transmission into reverse and then using the accelerator as a brake! Alternatively a retarder of the eddy-current or similar type could be used. Noise has often been advanced as a disadvantage of this type of engine but in fact there is no great difficulty in getting silence by conventional methods. Heat exchangers normally deal with the exhaust noise and the high-frequency intake noise which emanates from the compressor may be reduced to acceptable limits by a splitter type of silencer. For reduction of noise emanating from gears and casings, the expedient of under-bonnet insulation is generally adequate. Although much attention has been given to powering a passenger car with a gas turbine the author feels that the truck application is a simpler and probably more rewarding proposition. Both Boeing (10) and General Motors (8) have made such an application experimentally and the Austin 250 hp unit has been designed for the addition of a power turbine at a later date. The truck application is simpler than the passenger-car application for many reasons as it can make better use of high horsepower with its relatively high load factor, secondly there is more space for stowing a heat exchanger, and snap acceleration is less important. Finally a 250–300 hp diesel unit is quite an expensive item and makes the cost target for a turbine more attractive.

The turbine engine vehicle is a very much more attractive unit to drive than its reciprocating counterpart owing to its smoothness in torque and absence of heavy vibration. In addition, the performance of a turbine unit is superior to that of a reciprocating engine with a normal transmission of the same power, owing to its attractive torque characteristic. Stall torque is 2–2½ times full-load torque. This is not sufficient for a heavy vehicle but it much simplifies the transmission. Fig. 4, calculated for a 24-ton truck with a conservative assumption of a stall torque double the full-load torque, shows that a turbine with a three-speed transmission, preferably of the automatic type, gives a better performance than a diesel engine with five speeds. Both will, of course, have the same maximum speed but it will be noted that there are many areas of operation where the turbine shows considerable advantage, for instance it will be noted that a gradient of 1/45 could be climbed at a speed of 46 miles an hour instead of 38. The jumps between gears are much smaller and the 'engine' hangs on because it naturally gives more torque to meet higher loads while the reciprocating engine torque curve is not far from being flat, making gear changing more frequent.

CONCLUSIONS

Gas turbines have perhaps not made the progress that was initially hoped in the small turbine field. Nevertheless, they have fitted admirably into many markets particularly on the industrial side in cases where fuel consumption is not important and as aircraft auxiliary units where lightness is at a premium. In the automobile field advance in experimental units continues rapidly and one company has sold the first units for fire vehicle propulsion. Other firms have sold units to propel boats and light aircraft. It can therefore be comfortably assumed that the gas turbine will gradually become more and more widely used as development continues but that a sudden break-through is unlikely. Costs do not come down pro rata with size so it would seem that the small passenger car is going to present the greatest difficulty.

ACKNOWLEDGEMENTS

The author would like to acknowledge gratefully the co-operation he has received from many companies in the supply of information for his lecture, particularly that of Rolls-Royce for performance details of their 'Dart' compressor. He would like to record the great help he has received in the preparation of this lecture from Mr J. Barton, and from Dr R. J. Tonks and Mr M. C. Hathaway, and of the research team for their contribution to the work on which this lecture has been based, and in conclusion he would like to thank the Directors of the Austin Motor Co. for their permission to give this lecture.

APPENDIX I

Introduction

Component efficiencies and assumptions made in calculations for Figs 1–5.

(1) *Design point operation*

Compressor adiabatic efficiency	0·80
Compressor inlet temperature	288°K (518°R)
Compressor inlet pressure	14·7 lb/in² abs.
Turbine adiabatic efficiency	0·85
Combustion efficiency	0·98
Heat exchanger thermal ratio	0·80
Pressure loss in combustion chamber and ducts	5 per cent
Pressure loss in heat exchanger	4 per cent
Air leakage loss between compressor and turbine	3 per cent with heat exchanger 1 per cent without heat exchanger
Lower calorific value of fuel	10 300 C.H.U./lb (18 500 Btu/lb)

Compressors

Comparison of Austin 30 hp compressor and 250 hp compressor

Dimensions

	d_1	d_2	d_3	d_4	d_5
30 hp compressor	1·90	3·14	5·4	6·26	8·15
250 hp compressor	3·25	6·35	10·8	12·0	15·6

d_1 Eye-hub diameter, in.; d_2 Eye-tip diameter, in.; d_3 Impeller tip diameter, in.; d_4 Pitch circle diameter of diffuser leading edges, in.; d_5 Pitch circle diameter of trailing edges, in.

Air velocities at design operating point

	v_1	v_2	v_3
30 hp compressor	350	1142	966
250 hp compressor	400	1230	1102

v_1 Air approach velocity (axial) at intake, ft/s; v_2 Air velocity (absolute) at impeller tip, ft/s; v_3 Air approach velocity at leading edge of diffuser ring, ft/s.

Mach numbers at design operating point

	M_1	M_2	M_3	M_4
30 hp compressor	0·762	0·694	0·906	0·749
250 hp compressor	0·852	0·740	0·965	0·845

M_1 Eye-tip relative Mach No.; M_2 Eye-tip tangential Mach No.; M_3 Impeller tip Mach No.; M_4 Diffuser inlet Mach No.

Design parameters

	ϕ	R	d_2/d_3	m
30 hp compressor	0·266	3·0	0·582	0·87
250 hp compressor	0·292	3·5	0·588	4·7

$\phi = v_1/u$, Flow function; R Pressure ratio; m Mass flow rate, lb/s; u Impeller tip speed, ft/s.

Comparison of Rolls-Royce 'Dart' compressor and Austin 120 hp two-stage compressor (No. 8)

Dimensions

Compressor	First stage				
	d_1	d_2	d_3	d_4	d_5
Austin	2·437	5·750	12·000	12·922	16·500
Rolls-Royce	4·686	12·685	21·000	21·984	27·800
	Second stage				
	d_1	d_2	d_3	d_4	d_5
Austin	2·187	5·250	10·250	10·593	14·000
Rolls-Royce	3·625	9·900	17·600	18·312	23·800

d_1 Eye-hub diameter, in.; d_2 Eye-tip diameter, in.; d_3 Impeller tip diameter, in.; d_4 Pitch circle diameter of diffuser vane leading edges, in.; d_5 Pitch circle diameter of trailing edges, in.

Air velocities at design operating point

Compressor	First stage			Second stage		
	v_1	v_2	v_3	v_1	v_2	v_3
Austin	394	1030	950	260	874	852
Rolls-Royce	502	1180	1130	357	995	957

v_1 Air approach velocity (axial) at intake, ft/s; v_2 Air velocity (absolute) at impeller tip, ft/s; v_3 Air approach velocity at leading edge of diffuser vane, ft/s.

Mach numbers at design operating point

Compressor	First stage				Second stage			
	M_1	M_2	M_3	M_4	M_1	M_2	M_3	M_4
Austin	0·599	0·481	0·844	0·778	0·424	0·374	0·637	0·620
Rolls-Royce	0·890	0·760	0·858	0·822	0·541	0·474	0·652	0·625

M_1 Eye-tip relative Mach No.; M_2 Eye-tip tangential Mach No.; M_3 Impeller tip Mach No.; M_4 Diffuser inlet Mach No.

Design parameters

Compressor	First stage			Second stage		
	ϕ	R	d_2/d_3	ϕ	R	d_2/d_3
Austin	0·355	2·466	0·479	0·274	1·622	0·512
Rolls-Royce	0·365	3·380	0·604	0·310	1·805	0·562

ϕ Flow function, v_1/u; R Stage pressure ratio; m Mass flow rate, lb/s.

Heat exchangers for the 250 hp and 30 hp gas turbine engine

Design specifications.

	250 hp cross-flow		250 hp contra-flow				30 hp contra-flow	
	Air-side	Gas-side	Matrix passages Air	Matrix passages Gas	Side ducts Air	Side ducts Gas	Air	Gas
Fin thickness, in.	0·004	0·004	0·004		0·004		0·006	0·006
Plate thickness, in.	0·008	0·008	0·008		0·008		0·010	0·010
Height between plates, in.	0·052	0·150	0·100		0·100		0·072	0·210
Hydraulic diameter, in.	0·0343	0·054	0·038		0·137		0·050	0·0966
Fin pitch, fins/in.	24	24	32		3·85		16	13
Ratio of flow area/frontal area	0·202	0·6	0·794		0·877			
Total wetted surface area, ft²	1062	2005	1648		463		114·1	208·2
Number of passages	46	46	137	138	137	138	33	32
Passage length, in.	11·6	9·35	6·5		13·0	6·75	10	11·9
Passage width, in.	9·05	11·30	29·2		13·5	26·0		
Flow area, ft²	0·790	2·900	2·39		1·22	2·35	0·142	0·423
Height of matrix, in.	60·0	(10 in./unit)	29·7		29·7		9·65	9·65
Weight of matrix, lb	389	(64·9 lb/unit)	666					
Pressure drop, lb/in²	0·69	0·304	0·189	0·48	Included in matrix		0·6	0·2
Pressure drop, per cent	1·34	2·07	0·37	3·20			1·36	1·33
Pressure drop, per cent (Total)	3·41		3·57				2·69	
Thermal ratio	0·74		0·84				0·68	

APPENDIX II

REFERENCES

(1) WEAVING, J. H. 1957 *Amer. Soc. mech. Engrs*, Paper 57–GTP–2, 'Austin Vehicle Gas Turbine'.

(2) ANON. 1961 *Oil Eng.*, vol. 29, No. 377, p. 264, 'Automotive Power Plant Developments by Rover'.

(3) HUEBNER, G. J. 1957 *Soc. automot. Engrs, N.Y.*, vol. 65, p. 705, 'Automotive Gas Turbine Today and Tomorrow'.

(4) ANON. 1961 *Oil Eng.*, vol. 29, p. 337, 'Advanced American Design of 440 hp Turbine'.

(5) MALLINSON, D. N. and LEWIS, W. G. E. 1948 *Proc. Instn mech. Engrs, Lond.*, vol. 159, p. 198, 'Part Load Performance of Various Gas Turbine Engine Systems'.

(6) SWATMAN, I. M. and MALOHN, D. 1960 *Soc. automot. Engrs, N.Y.*, Summer Meeting 1960 No. 187A, 'Advanced Automotive Gas Turbine Engine Concept'.

(7) SWATMAN, I. M. 1961 *Soc. automot. Engrs, N.Y.*, *Int. Congr.*, Paper No. 219A, 'Development of the Ford 704 Gas Turbine Engine'.

(8) GUERNSEY, R. W. 1961 *Soc. automot. Engrs, N.Y.*, Summer Meeting 1961, Paper No. 383A, 'Field Experience for GMT–305 Gas Turbine in Military Applications'.

(9) TURUNEN, W. A. and COLLMAN, J. S. 1959 *Soc. automot. Engrs, N.Y.*, Paper No. 24T, 'GT.305 Regenerative Engine in Firebird III'.

(10) STODDARD, R. L. and HARRISON, R. H. 1959 *Soc. automot. Engrs, N.Y.*, Paper No. 217, 'Installation Status of Gas Turbine Engine'.

(11) ANON. 1960 *Oil Eng.*, vol. 27, No. 318, p. 384, 'Extension of American Powered Range Boeing 520 Series'.

(12) ANON. 1945 *Oil Eng.*, vol. 22, No. 256, p. 236, 'Standardized Arrangement of Small Gas Turbines'.
1955 *Oil Eng.*, vol. 23, No. 269, p. 274, 'The "Paloust" Air Compressor Engine'.

(13) CARELLI, A. 1957 *Amer. Soc. mech. Engrs, Mech. Engng*, July, p. 645, 'Development of a Gas Turbine for Road Vehicles'.

(14) ANON. 1961 *Oil Eng.*, April, vol. 28, No. 330, p. 465, 'Introducing a British 250 hp Design' (First Austin Industrial Turbine).

(15) POPE, A. W. 1958 *Proc. Instn mech. Engrs, Lond.*, vol. 172, p. 301, 'Design and Development of Four Light-weight High-speed Marine Gas Turbines for Electric Generator Drive'.

(16) ANON. 1957 *Oil Eng.*, September, vol. 25, No. 289, p. 194, 'The Second Ruston Gas Turbine'.

(17) ANON. 1957 *Oil Eng.*, March, vol. 24, No. 283, p. 431, 'A New British Small Gas Turbine for Industrial Duty'.

(18) WOSIKA, L. F. 1952 *Trans. Amer. Soc. mech. Engrs*, vol. 74, p. 1337, 'Radial Flow Compressor and Turbines for the Simple Small Gas Turbine'.

(19) ANON. 1956 *Oil Eng.*, October, vol. 24, No. 278, p. 237, 'Versatile Range of Small Gas Turbines'.

(20) ANON. 1954 *Oil Eng.*, vol. 21, No. 250 and vol. 22, No. 251, 'A British 60 bhp Engine, Rover IS/60'.
1960 *Oil Eng.*, vol. 28, No. 322, p. 150, 'Small Power Unit Developments, Rover IS/90'.

(21) ANON. 1960 *Oil Eng.*, vol. 28, No. 320, p. 3307, 'Developed Design for Improved Performance (Budworth "Brill" Mk. 2)'.

(22) ANON. 1960 *Oil Eng.*, vol. 28, No. 320, p. 76, 'For Portable Intermittent Duty (BM.6002)'.

(23) ANON. 1954 *Oil Eng.*, vol. 22, p. 78, 'Light-weight Design for Fire Fighting Duty'.

(24) MINISTRY OF AVIATION, *D.I.G.T. Rep.* X–A–2.

(25) BALJE. 1952 *Trans. Amer. Soc. mech. Engrs*, vol. 74, p. 451.

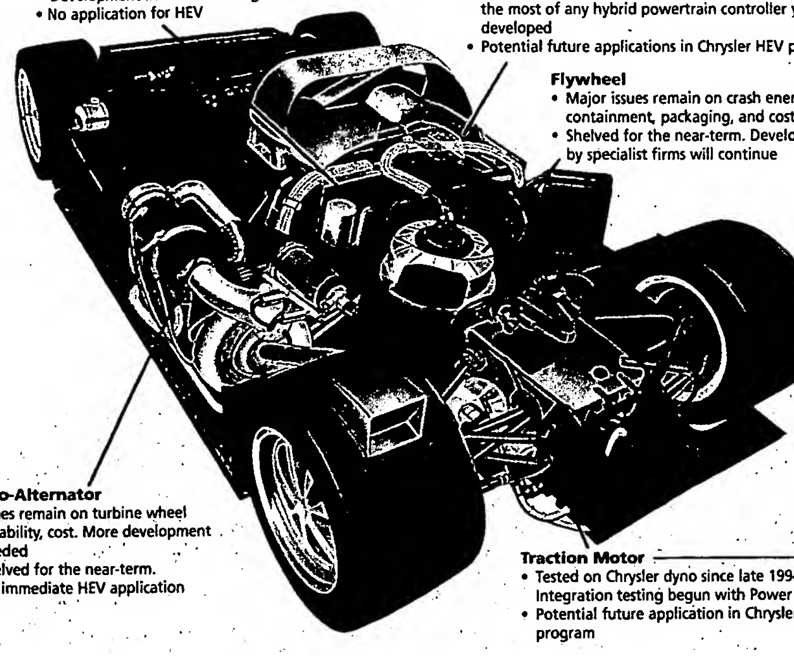
(26) ANON. 1960 *Oil Eng.*, vol. 27, No. 318, p. 386.

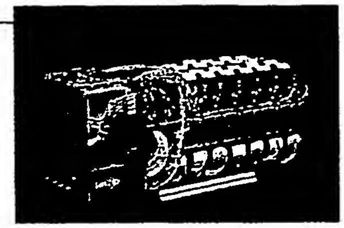
(27) MALLINSON, D. N. and LEWIS, W. G. E. 1948 *Proc. Instn mech. Engrs, Lond.*, vol. 159, p. 311, 'Small Axial Flow Compressors'.

(28) MINISTRY OF AVIATION 1960 *D.I.G.T. Rep.* 20–R–6, November.

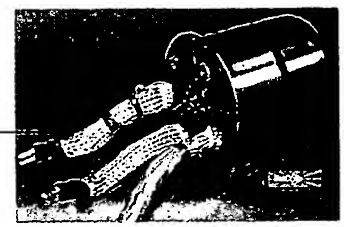
(29) MINISTRY OF AVIATION 1956 *D.I.G.T. Rep.* 4–A–1, February.

MADE AND PRINTED IN GREAT BRITAIN BY WILLIAM CLOWES AND SONS, LIMITED, LONDON AND BECCLES





Patriot race car spawned valuable cutting-edge technology and over 60 patents in just three years of development. Program suppliers elude to an unconfirmed "bargain" $33 million investment.



DNF for Patriot

By Gerry Kobe

Chrysler transfers Patriot technologies to its Hybrid Electric Vehicle (HEV) program.

No one could accuse Chrysler of "aiming too low" in 1993, when it began plans for a 200+ mph flywheel/turbo alternator hybrid race car. Dubbed "Patriot," the vehicle proceeded on the challenging premise that the breakthrough technologies needed to make it viable, would happen on-schedule, and function as planned.

Last month, Chrysler pulled the plug on Patriot.

According to Tom Kizer, executive engineer-Liberty, two important technologies held the program back. The first is a reliable, ceramic turbine-wheel, which he says "is as far from reality today as it was when the project started." And the second is the durability and safety aspects of the flywheel, which VP-engineering Francois Castaing attributes Chrysler "setting the bar so high in terms of energy-density."

However, key components of Patriot, as well as some 60 patented technologies will transfer to Chrysler's HEV program.

One example is Patriot's traction motor, which spent over a year on the dyno. Tested to about 55% of design capability, its unique architecture and cooling method give it outstanding performance characteristics. Kizer says if this design were equal in weight to his company's NS electric minivan induction motor, performance would improve by a factor of four.

Additionally, Patriot's power controller combines Insulated Gate Bipolar Transistors (IGBTs) with new materials that let the less-than-five-ounce modules carry 600 amps at up to 1,700 volts. Conventional modules are seven times heavier. Creative packaging of the modules in a "window frame" construction also allows coolant to bleed off 80 kW of waste heat for reliable operation.

Even a new DC busbar contributes to a smaller, lighter, cooler electronics package. It offers 1,000 amp capacity in a seven pound three ounce unit. A commercial 1,000 amp busbar weighs 70 lbs!

These and many more Patriot-inspired technologies will surface in Chrysler's "series-diesel hybrid" program headed by Steve Speth. According to Speth, Detroit Diesel is already signed on to supply a 40-60 hp, lightweight, direct-injection diesel. TRW will provide electronic power steering, ITT has electric brakes and JCI/Bolder supplies spiral-wound lead acid batteries.

The HEV program timing is about four years, with Catia packaging work in progress now through fall. At that point, component type and size will be spec'd. About July '97, design and procurement of Generation 1 parts begins, which will be built into an aluminum-intensive Neon. From there, dyno work begins to correlate simulations, leading to Generation 2 parts in June 1998. Those parts will represent what Chrysler thinks it can put into production vehicles 10-15 years down the road.

Patriot's contribution? Chrysler thinks it may have shaved a year to 18 months off the HEV timeline. AI